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IN THIS ISSUE

BEFORE Reynolds' theory of hydrodynamic lubrication was known, the performance of journal bearings was far superior to that of thrust bearings, because journal bearings develop a wedge-shaped oil film by assuming an eccentric position under load, while the geometry of the thrust bearings then existing did not favour the formation of such a wedge. Following Reynolds' theory, however, the very efficient hydrodynamic thrust bearings of Kingsbury and Michell were developed; but since then, no practical thrust bearings based on new hydrodynamic principles were forthcoming.

It is still considered necessary that the surfaces of a bearing should converge in the direction of motion of the moving bearing element, so as to produce a fluid wedge in order that positive pressure may be developed under steady-state conditions. Consequently, hydrodynamic bearings usually have one of the following features: (1) The journal centre is moved into an eccentric position with respect to the bearing centre, as in the case of a full, partial, or fitted journal bearing; (2) depressions, tapered in the direction of motion, are machined into the fixed bearing surface as in the case of the so-called fixed-pad bearing; (3) the stationary bearing element is mounted upon a pivot, enabling it to adjust its position freely and, thus, form the desired wedge, as in the Kingsbury or Michell bearing.

It was recently discovered that the fitted spherical bearing called "Hydrosphere" is capable of hydrodynamically supporting loads of very great magnitude. This bearing is described in the article on page 327 and the results of several experimental tests are analysed with respect to the qualitative theory presented.

Based on principles akin to those of the Hydrosphere, the "Floating Bearing," developed in France, is claimed to be a practical proposition for many applications. In this bearing, the shaft is held in a central position by the application of a pressure field, i.e., the shaft is supported entirely by fluid pressure. A short description of this bearing is given on page 336. It should be noted that oil is not the only suitable working medium, but that, for instance, air can also be used. One question which has not been elucidated is that of sealing the bearings, and this, in our opinion, must be fraught with difficulties.

* * *

The three-phase motor with a rotor-fed commutator has become a standard machine in the electrical industry. However, being a comparatively recent development, its operation and possibilities are not generally known, although this machine offers particularly simple solutions to a number of industrial problems, such as the speed control of electrically-operated machine tools. The article on page 330 reviews the principles on which the shunt-commutator motor is based and refers to some interesting industrial applications.

* * *

By "shop-run tolerances" is understood a general system of limits within which work can be produced economically. These tolerances apply to dimensions and conditions for which limits or special instructions are not specified on the drawing.

While an earlier article (see *The Engineers' Digest*, August, 1948, pp. 271-273) deals with such tolerances for iron castings, blanked and pierced parts, and parts bent on disks and bending machines, the article on page 350 of the present issue gives data for components produced on screw machines and general machine shop equipment. In particular, eccentricity and angularity tolerances for parts threaded on screw machines are discussed. Various tables should be of special practical value, as they reflect the practice of the General Electric Company, with whom the author is associated.

* * *

In the article "Carbide Drills" on page 353 our attention is drawn to the great advantages these tools afford in comparison with drills made of ordinary tool steel or even of high-speed steel, i.e., heavier feeds, longer drill life, and greater accuracy of holes. It is not our intention to deny these merits, but we think it should be pointed out that, owing to their high cost, carbide drills are only economical in use if the superior properties of these tools can be fully exploited. Unfortunately, this aspect, obvious as it may be, is sometimes overlooked in practice. For, what is the object of employing expensive tools on old and insufficiently powerful machines? It should be concluded, therefore, that carbide drills are useful only when machines are available which permit full utilization of the speeds and feeds of which these drills are capable.

* * *

The application of stress theorems to practical design problems cannot be regarded as a strictly scientific method of providing the data required for dimensioning machine parts. The stress distribution, even in a simple component, is so complex that only stress analysis by practical means, such as the application of strain gauges, photo-elasticity, and the brittle lacquer method, can lead to an exact solution of the problem.

The designer can only hope to obtain an approximate picture of the stress maxima occurring and is compelled to base his calculations on a number of assumptions and generalisations, which may approach, but never coincide, with actual conditions. However, in most cases it will be found that calculation can provide the necessary data if the designer logically examines the problem with which he is confronted. An example of this is given in the article on page 337 which should prove to be a useful guide to the solution of stress problems in centrifugal pumps.

U. S. A.

The Use of Oxygen in Steelmaking

By G. V. SLOTTMAN. (From *The Engineers' Digest, American Edition*, Vol. 5, No. 5, July, 1948, pp. 223, 268.)

THE use of oxygen in metallurgical processes is a subject which, like the weather, has often been discussed. Until recently, however, very little of an engineering nature has been accomplished either in the regulation of atmospheric precipitation or in the practical application of oxygen in the combustion and refining reactions common to the metals producing industries.

The theoretical advantages of high flame temperature, of high calorific intensity, and of low flue gas volume associated with the oxygen or oxygenated air combustion of fuels could have been deduced from Priestley's first experiments with a glowing taper. The depressing role that nitrogen plays in air-fuel combustions has been long recognized by combustion engineers. Similarly, in the oxidation reactions of metals refining, the use of air or metal oxides as sources of oxygen frequently involves depressing heat effects which retard the reactions, imposing limitations on output and on the type of raw materials which can be processed.

As early as 1924, the general principles underlying the use of oxygen in the steel industry were discussed in detail at a symposium held under the auspices of the Bureau of Mines and, during the past decade, some experiments were made in Germany using oxygenated air in the blast furnace and in the Bessemer. The open hearth furnace, which is by far the major steel producing unit in the American industry, was given little attention since it was felt that oxygenated air combustions would have a serious effect on the refractories. The use of gaseous oxygen as a bath reagent had been considered, but techniques for applying the large volumes required had not been developed and, moreover, it was generally felt to be too expensive an oxidant compared with iron ore.

The open hearth process for making steel, while being capable of closely controlling product quality and of processing large volumes of scrap metal, is greatly inferior to the Bessemer process with regard to thermal efficiency and production rate. Scrap melting, particularly in older furnaces, is very slow, and the refining period, involving the oxidation of carbon by an oxidizing slag, is prolonged because of the slow rate of transfer of oxygen from the slag to the metal.

In 1946, two developments involving the melting of cold materials and their subsequent refinement by the oxidation of impurities were made in the open hearth practice which will bear heavily in the future thinking not only of steel plant technologists but of those engineers concerned with metal refining operations. Engineering answers were found to the two problems of developing an oxygen fuel flame which would not affect the furnace refractories, and of injecting large volumes of oxygen into the molten bath.

As regards the combustion use, previous thinking had been in terms of oxygenated air. Since the open hearth flame develops from the diffusion of the fuel into an atmosphere of preheated air, the effect of oxygenated air is to increase temperatures generally throughout the furnace without achieving any control over flame length and the location of maximum flame intensity. By contacting the fuel stream with a stream of relatively high purity oxygen and allowing the partly burned gases to diffuse into the normal atmosphere of preheated air from the regenerators, a flame was developed which could be accurately controlled in shape, and which could release considerably larger quantities of heat in a given combustion space.

Coincident with the application of application of oxygen in combustion, means were developed for injecting oxygen directly into the molten bath. Since steel burns in an oxygen atmosphere when heated to a red heat, and since refractory tubes are expensive, cumbersome and sensitive to thermal shock, oxygen injection was considered to involve a difficult problem in equipment. The answer to this problem was found to be simplicity itself. When oxygen is passed through a bare steel pipe at sufficient velocity, the cooling effect of the gas on the pipe reduces the melt-off rate of the pipe to the point where it becomes practical to inject large volumes of oxygen directly into the molten steel bath with a simple device which can be readily handled on the busy charging floor.

Recognizing the significance of these two developments with regard to the steel productive capacity of the country, the steel industry immediately embarked on a large-scale experimental program to determine the economics involved in the use of oxygen in combustion and as a bath reagent. No development has ever been tested by any industry on such a scale and in such a comparatively short space of time. From the tremendous amount of data accumulated, conclusions were quickly drawn with reference to the present and future role of oxygen in steelmaking.

The combustion use, requiring between 400-800 cubic feet of oxygen per ton of steel produced, or some 2,000,000 tons of oxygen per year for all open hearth furnaces as against an oxygen producing capacity for the country of some 750,000 tons per year, was deemed to be largely a matter for the future. The economics of this operation, applied to large modern furnaces, indicates that oxygen must be available at relatively low cost. While the value of oxygen in small furnaces of older vintage is considerably higher, any such uses are largely incidental to the main volume of steel production. Combustion uses of oxygen in other industries will similarly involve large volumes of low cost oxygen, and will be dependent on the development of the " tonnage " or low purity oxygen plant which is presently proceeding at a moderate pace.

Oxygen as a bath reagent, using present sources of supply, is economical in the manufacture of certain grades of steel, and has been in commercial use for more than a year. Low carbon stainless steels, transformer steels and ingot iron, which normally require long refining times under highly oxidizing slags and at high furnace temperatures, are typical products in whose production the use of gaseous oxygen has become a standard practice. Since these products represent only a small portion of the total steel production, the effect of oxygen in increasing overall steel capacity has been relatively small.

A path has been broken, however, in what promises to be a new field in metal refining. The reaction of metal impurities with solid oxidants generally involves heat effects which range from the mildly exothermic to the strongly endothermic, and which are often limited in speed by the rate at which heat can be introduced into the metal-slag system. Frequently, the solid oxides formed remain as contaminations in the metal product, and the rate of evolution of gaseous products is not sufficient to prevent absorption by the metal of the heating gas constituents. In steelmaking, the highly exothermic nature of the oxidation reactions with gaseous oxygen, and the rapid rate of carbon monoxide evolution from the molten bath, have facilitated the production of superior quality steels, at high production rates in furnaces of the electric and open hearth types.

A number of other uses of oxygen as a reagent were also investigated by the steel industry. These include the control of blast furnace iron analysis with respect

(*Continued on page* 358).

The Hydrosphere—A New Hydrodynamic Bearing

By M. C. SHAW and C. D. STRANG, Jr. (From *Journal of Applied Mechanics*, Vol. 15, No. 2, June, 1948, pp. 137-145, 16 illustrations.)

IT was recently discovered that the fitted spherical bearing called the " hydrosphere " is capable of hydrodynamically supporting loads of very great magnitude. The two components of the bearing are fitted together without clearance, the journal being a portion of a perfect sphere. The bearing surface is divided into a number of sectors by equally spaced radial oil grooves. Under axial loading, there is no provision for the formation of a wedge in the direction of motion of the moving surface; hence the source of the hydrodynamic pressure which is experimentally found to develop in this bearing is not readily apparent from the ordinary conceptions of lubrication.

THEORY

In the hydrosphere the crescent-shaped space required for the oil film is obtained by having the centre of curvature of the journal move in an axial direction away from the centre of curvature of the bearing surface (see Fig. 1). Thus, while this bearing is without a wedge in the direction of motion of the moving surface a wedge does exist perpendicular to the direction of motion.

In a first approximation, the hydrosphere can be regarded as equivalent to a slider bearing with transverse curvature (see Fig. 2). Reynolds' equation is still valid when the film thickness h varies normal to, as well as in the direction of, the moving surface, hence we may write

$$\frac{\partial}{\partial x}\left(\frac{h^3}{\mu}\frac{\partial p}{\partial x}\right)+\frac{\partial}{\partial z}\left(\frac{h^3}{\mu}\frac{dp}{\partial z}\right)=6\,\frac{\partial (hU)}{\partial x}, \quad .. \quad (1)$$

where $h = h(x, z)$ = oil thickness [in]; μ = viscosity [reyns]; p = pressure [psi], U = velocity of the moving surface [ips], the x- and z-co-ordinates being as shown in Fig. 2. The variation of pressure in the x- and z-directions must be somewhat as shown in Fig. 3 for any bath-lubricated slider bearing with zero pressure along all the boundaries.

If the viscosity of the lubricant, as well as the film thickness, is constant at all points, eq. (1) reduces to Laplace's equation:

$$\frac{\partial^2 p}{\partial x^2}+\frac{\partial^2 p}{\partial z^2}=0 \quad .. \quad .. \quad (2)$$

If in a parallel-surface bearing of this type a pressure gradient such as that shown in Fig. 3 were developed in the x-direction, then the pressure gradient in the z-direction would have to be as shown in Fig. 4 to satisfy eq. (2). This obviously cannot be obtained physically. The only other possible pressure distribution satisfying eq. (2) is one in which the pressure is constant at all points, and hence zero at all points if the pressures at the boundaries are zero. Thus, Reynolds' equation predicts a zero-load capacity for a parallel-surface bearing.

Equation (1) can be applied to a bearing of the Kingsbury or Michell type, in which h varies linearly in the x-direction only, with the viscosity assumed constant throughout the oil film, so that

$$\frac{\partial^2 p}{\partial x^2}+\frac{\partial^2 p}{dz^2}=\frac{3}{h}\frac{\partial h}{\partial x}\left(\frac{2\mu U}{h^2}-\frac{\partial p}{\partial x}\right) \quad .. \quad (3)$$

Unlike Laplace's equation, eq. (3) is not satisfied when the pressure is constant at all points. It is, however, satisfied by a pressure distribution such as that of Fig. 3, and approximate solutions have in fact been determined.

Applying the same general method to a bearing of the type shown in Fig. 2 in which the film varies in the

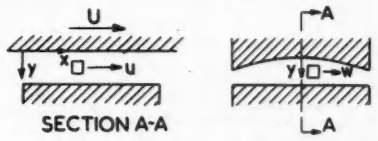


Fig. 2. **Slider bearing with transverse curvature.**

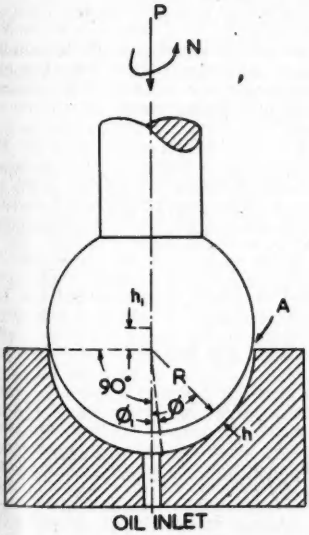


Fig. 1. **The hydrosphere showing crescent-shaped oil film.**

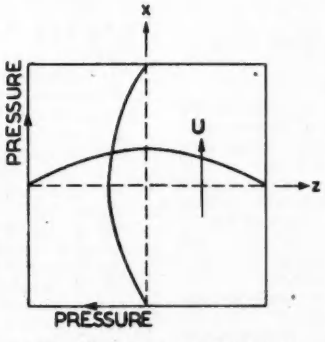


Fig. 3. **Plan view of bath-lubricated slider bearing, showing approximate pressure variation.**

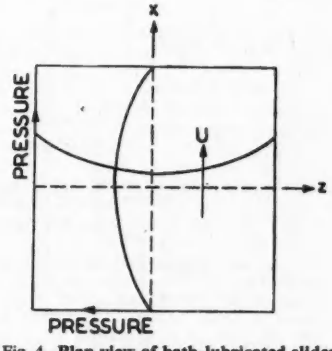


Fig. 4. **Plan view of bath-lubricated slider bearing showing approximate pressure variation required to satisfy Laplace's equation.**

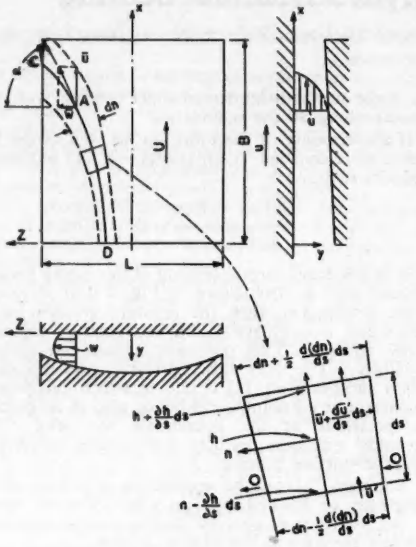


Fig. 5. **Slider bearing with transverse curvature, showing elementary portion of mean stream tube.**

z–direction only, eq. (1) reduces to

$$\frac{\partial^2 p}{\partial x^2} + \frac{\partial^2 p}{\partial z^2} = -\frac{3}{h}\frac{\partial h}{\partial z}\frac{\partial p}{\partial z} \qquad .. \qquad (4)$$

Equation (4) is satisfied not only when pressure is constant at all points, but also, under certain conditions, by a pressure distribution of the type shown in Fig. 3. However, as eq. (4) does not explicitly contain the variables U and μ which determine the development of pressure, it will not reveal whether the bearing is capable of operating hydrodynamically, unless a complete solution of eq. (4) is obtained, and this is a task at least as laborious as that of solving eq. (3).

Another attempt can be made to find a solution for the bearing in Fig. 2 by introducing instead of fixed Cartesian co-ordinates, a moving pair of rectangular co-ordinates, directed along, and normal to, a streamline, thus forming an s–n system.

In the slider bearing of Fig. 5, the velocity variation across the oil film at point A will be somewhat as shown in the end and side elevations, respectively. The mean streamline is defined as a line tangent to the resultant of the mean velocity vectors $\bar{u} = \bar{u}(x)$ and $\bar{w} = \bar{w}(z)$ at all points along its length. A mean stream tube can be formed by erecting imaginary vertical walls along the two dotted streamlines adjacent to the mean streamline CAD, and extended completely across the oil film, the net flow across the vertical walls of the mean stream tube then being equal to zero.

In the plan view of an element of the mean stream tube also shown in Fig. 5, the mean velocity along the stream tube is denoted by $\bar{u}'$. In accordance with the law of conservation of matter, the net mass of fluid leaving the quasi-rectangular elementary volume must be zero, or

$$\left[\left(\rho + \frac{1}{2}\frac{\partial \rho}{\partial s}\,ds\right)\left(\bar{u}' + \frac{1}{2}\frac{d\bar{u}'}{ds}\,ds\right)\times \left(dn + \frac{1}{2}\frac{d(dn)}{ds}\,ds\right)\left(h + \frac{1}{2}\frac{\partial h}{\partial s}\,ds\right)\right] - \left[\left(\rho - \frac{1}{2}\frac{\partial \rho}{\partial s}\,ds\right)\left(\bar{u} - \frac{1}{2}\frac{d\bar{u}'}{ds}\,ds\right)\times \left(dn - \frac{1}{2}\frac{d(dn)}{ds}\,ds\right)\left(h - \frac{1}{2}\frac{\partial h}{\partial s}\,ds\right)\right] = 0, \quad (5)$$

where ρ = mass density of the lubricant. When higher-order terms are ignored, this continuity relationship becomes

$$\partial(\rho h\,\bar{u}'\,dn)/\partial s = 0. \qquad .. \qquad .. \qquad (6)$$

Fig. 6 shows a developed sectional view of the bearing, taken along the mean streamline CAD. Since s is not in the direction of motion, the velocity of the moving surface will be equal to U sin α, where

$$\alpha = \tan^{-1}(\bar{u}/\bar{w}) \qquad .. \qquad .. \qquad (7)$$

The pressure and shear forces acting in the s–direction are shown on an enlarged particle, ds, dy, dn, in Fig. 6. If the inertia forces are neglected, then for static equilibrium the sum of all the forces in the s–direction must be zero:

$$\left[\left(p - \frac{1}{2}\frac{\partial p}{\partial s}\,ds\right)dy\left(dn - \frac{1}{2}\frac{d(dn)}{ds}\,ds\right)\right] - \left[\left(p + \frac{1}{2}\frac{\partial p}{\partial s}\,ds\right)dy\left(dn + \frac{1}{2}\frac{d(dn)}{ds}\,ds\right)\right] + \left(\tau_s + \frac{1}{2}\frac{\partial \tau_s}{\partial y}\,dy\right)ds\,dn - \left(\tau_s - \frac{1}{2}\frac{\partial \tau_s}{\partial y}\,dy\right)ds\,dn + p\,ds\,dn\,\frac{d(dn)}{ds} = 0 \quad (8)$$

or

$$\partial p/\partial s = \partial \tau_s/\partial y \qquad .. \qquad .. \qquad (9)$$

where τ_s is the shear stress in the s–direction. By Newton's law for viscous shear

$$\tau_s = \mu\,\partial u'/\partial y \qquad .. \qquad .. \qquad (10)$$

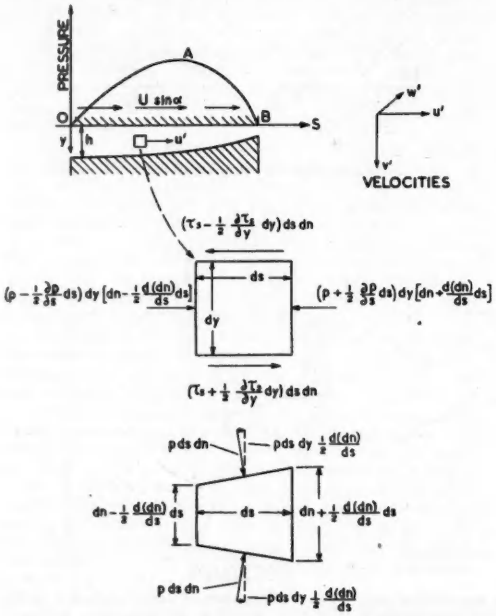


Fig. 6. **Developed sectional view of bearing in Fig. 5, taken along mean streamline CAD.**

and hence

$$\frac{\partial^2 u'}{\partial y^2} = \frac{1}{\mu}\frac{\partial p}{\partial s} \quad .. \quad .. \quad (11)$$

As a consequence of the thinness of the film, $\partial p/\partial s$ may be regarded as independent of y. Integrating eq. (11) twice with respect to y, and determining the integration constants from the conditions $u' = U \sin \alpha$ when $y = 0$, and $u' = 0$ when $y = h$, gives

$$u' = \frac{1}{2\mu}\frac{\partial p}{\partial s}\, y\,(y - h) + \frac{h-y}{h} U \sin\alpha, \quad (12)$$

and hence

$$\bar{u'} = \frac{1}{h}\int_0^h u'\, dy = -\frac{h^2}{12\mu}\frac{\partial p}{\partial s} + \frac{U \sin\alpha}{2} \quad .. \quad (13)$$

From eqs. (6) and (13)

$$\partial\left(\frac{\rho h^3}{12\mu}\frac{\partial p}{\partial s}\, dn\right)\Big/\partial s = \partial\left(\rho\,\frac{U}{2}\sin\alpha\; h\, dn\right)\Big/\partial s \quad (14)$$

The quantities in parentheses, in eq. (14), both represent the rate of mass flow along the stream tube. Eq. (14) states that the mass flow due to the pressure gradient $\partial p/\partial s$ is equal and opposite to the shear flow in moving from point to point along the mean stream tube. When $\partial p/\partial s > 0$ the flow will be in the negative s-direction, while the shear flow due to the motion of the bearing surface will always be in the positive s-direction.

The bearing will function hydrodynamically if the pressure gradient (see Fig. 6) is such as to cause a pressure flow in the negative s-direction from O to A, and in the positive s-direction from A to B. In order that the total flow at each section may remain constant, the shear flow must then decrease continuously from O to B. Hence positive pressure will be developed in any bearing provided the quantity $\rho U \sin\alpha\; h dn$ decreases continuously along a mean stream tube.

The film thickness h in eq. (14) may be any function of x and z. For the bearing in Fig. 2, U is independent of s, and h decreases in the s- direction; the angle α will also decrease in the s-direction as a consequence of $\bar{w}$ increasing in the z-direction in accordance with the continuity relationship. If the decrease in density along the stream tube be assumed to compensate any divergence of the streamlines, then the bearing in Fig. 2 will act hydrodynamically. Thus, a bearing with a wedge normal to the direction of motion of the slider, but without a wedge in the direction of motion, can function hydrodynamically if the lubricant has a velocity component in the normal direction.

The popular method of causing the shear flow $\rho U \sin\alpha\; h dn$ to decrease continuously along a mean stream tube involves the use of a wedge-shaped oil film in the direction of motion. FOGG has recently shown that a bearing may function hydrodynamically without an oil wedge as a consequence of the decrease in density in the direction of motion accompanying the rise in oil temperature. It is now evident that other sources of pressure exist in a bearing, such as the following:—(1) any forces (for example, centrifugal inertia forces) tending to increase flow perpendicular to the direction of motion of the moving bearing element, so as to alter the path of a mean streamline and hence the angle α; (2) any factors (such as the geometry of the bearing) which cause the mean streamlines to converge; (3) changes in density due to changes in pressure in the oil film; (4) a wedge normal to the direction of motion of the moving bearing element.

Eq. (14) is directly applicable to the hydrosphere after making appropriate substitutions. The difference in curvature between the oil films of the bearing in Fig. 5 and the hydrosphere is negligible in view of the thinness of these films. From Fig. 1 it is evident that the film thickness at any point on the bearing is

$$h = h_1 \cos\phi - R + \sqrt{R^2 - h_1^2 \sin^2\phi} \quad (15)$$

or $$h \cong h_1 \cos\phi \quad .. \quad .. \quad .. \quad .. \quad .. \quad (16)$$

where h_1 = maximum film thickness or axial "eccentricity" of the spherical journal. The velocity of a point on the journal surface is

$$U = R\,\omega \sin\phi \quad .. \quad .. \quad .. \quad (17)$$

where R = radius of the bearing in inches, and ω = angular velocity of journal; hence from eqs. (14), (16), and (17)

$$\frac{\partial}{\partial s}\left(\frac{p\,(h_1\cos\phi)^3}{12\,\mu}\frac{\partial p}{\partial s}\, dn\right) = \frac{h_1 R\omega}{4} \times \frac{\partial\,(p \sin 2\phi \sin\alpha\, dn)}{\partial s} \quad .. \quad (18)$$

Fig. 7 shows a single pad of the hydrosphere with a mean streamline for this bearing and the manner in which α is measured.

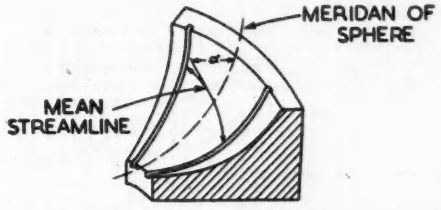


Fig. 7. Single pad of hydrosphere showing mean streamline and manner in which α is measured.

Eq. (18) shows that positive pressure will be developed only over the portion of the hydrosphere for which the righthand side of the equation is negative. To determine the useful portion of the bearing surface, the paths of the streamlines must be known. The width of the stream tube will not vary greatly for the bearing in Fig. 5, but for the hydrosphere it will be assumed that dn varies approximately as $\sin\phi$, since the width of the stream tube for pure radial flow between concentric spheres also varies as $\sin\phi$. Disregarding the effect of changes in ρ and α on the right-hand side of eq. (18) relative to the divergence of the streamlines and changes in surface velocity and film thickness, the sign of the righthand side of eq. (18), to a rough approximation, will be that of the quantity

$$\partial\,(\sin 2\phi \sin\phi)/\partial s = (\sin 2\phi \cos\phi + 2 \sin\phi \cos 2\phi)\,\partial\phi/\partial s \quad (19)$$

where $$\partial\phi/\partial s = \cos\alpha/R \quad .. \quad .. \quad .. \quad (20)$$

Thus $\partial\,(\sin 2\phi \sin\phi)/\partial s$ is negative for values of ϕ greater than $\tan^{-1}\sqrt{2} \simeq 55$ deg. Therefore, positive pressure will only develop in the upper portion of the hydrosphere, and this is indeed fortunate, for thus the hydrosphere need not be limited to dead-end applications, but the shaft may be extended through the bottom portion of the bearing.

An analytical determination of the pressure distribution cannot be obtained unless the manner in which ρ, α, and dn, vary with s is determined; and as, for instance, the value of α alone is influenced by all quantities affecting the flow from the bearing in a radial direction, such as pressure, shape of film in radial direction, and centrifugal inertia force on the oil, the discussion of the hydrosphere will now be continued on an empirical basis.

(To be continued).

Design and Industrial Applications of the Shunt-Commutator Motor

By M. MELONE. (From *L'Elettrotecnica*, Vol. 35, No. 4bis, April, 1948, pp. 166-182, 34 illustrations.)

1. INTRODUCTION

THE usual asynchronous three-phase motor has the advantage that it can maintain a nearly constant running speed when the load torque is varied; this is usually known as a " shunt " characteristic, because it is typical of the shunt-wound d.c. motor. This advantage can become a disadvantage in cases where the motor speed has to be varied, for instance, when operating machine tools which have to be adjusted to the optimum working speed.

The most primitive method of speed control consists in the insertion of a resistance in the rotor circuit, but this has the following disadvantages:—(1) the motor efficiency decreases; (2) speed regulation at zero load is impossible; (3) speed instability is considerable, since, for a given value of the inserted resistance, speed depends only on the rotor current, that is, on the load, as shown in Fig. 1. In this figure, the

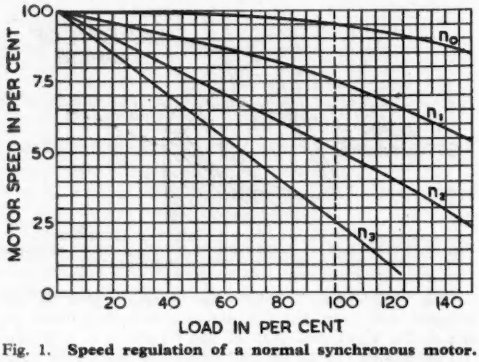


Fig. 1. **Speed regulation of a normal synchronous motor.**

curve n_0 is the normal curve of the asynchronous motor with the rotor winding in short-circuit, the speed remaining nearly constant when the load is varied, while curves n_1, n_2 and n_3 show the variation of speed when various additional resistances are inserted. The effect of the rotor losses decreases with reduced loads, and slip also decreases; the four curves thus have the same origin. With increased speed control, the curves differ more and more from the ideal " shunt " characteristic. This was not, therefore, a satisfactory solution, and it was only when three-phase commutator motors appeared that the problem of providing speed control was solved for the asynchronous motor without loss of energy.

Two main types of commutator machines were then developed:—(1) stator and rotor-fed motors, using a tapping transformer, with a step-by-step regulation, which was then made continuous by means of an induction regulator; (2) rotor-fed machines, better known as Schrage motors, in which speed control above and below the synchronous speed is a continuous process, not requiring any intermediate adjustment. This article deals exclusively with the Schrage type rotor-fed commutator motor, which is suitable for a great number of industrial applications.

2. PRINCIPLES OF OPERATION

The operation of the Schrage motor is similar in principle to that of the usual three-phase asynchronous motor, but, in addition, speed can be regulated simply by shifting the brushgear on the commutator, thus introducing into the secondary circuit an additional

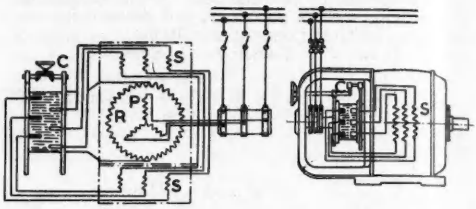


Fig. 2. **Schematic diagram of three-phase commutator motor with shunt characteristic.**

voltage of suitable magnitude and phase which alters the running speed.

The three-phase Schrage motor consists mainly of a pack of laminations constituting the stator, and of a rotor with three different windings; it also has a commutator with adjustable brushes and three slip-rings, as shown in Fig. 2. The primary or supply winding P is accommodated within the rotor and is directly connected to the supply net-work by the three normal slip-rings. In the outer part of the rotor, that is, in the vicinity of the air-gap, there is a second auxiliary or regulator winding R provided with a commutator, similarly to an ordinary winding of a d.c. machine. A special device is used to alter the position of the brushes. Two lines of brushes are used, their number being equal to twice the number of pole pairs per phase of the secondary winding.

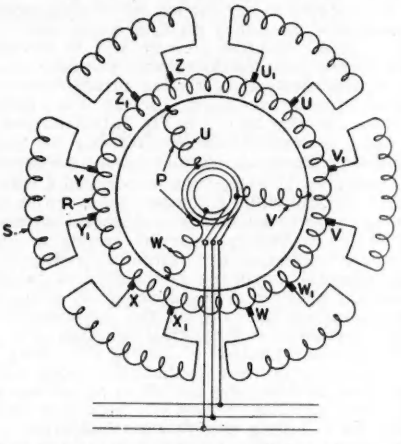


Fig. 3. **Diagram showing the windings of a Schrage motor.**

The secondary winding S, accommodated in the stator, is of the open polyphase type normally used in a.c. machines; the ends of each phase are connected to the commutator brushes as shown in Fig. 3 for a two-pole machine with a six-phase secondary S.

2A. SPEED CONTROL

When the primary winding P is fed from the supply network, a rotating field is generated which rotates at a speed $n_c = 60 f_1/p$ (rpm).

The following symbols will be used:—

n_c = rpm of the rotating field and synchronous speed of the motor; n = rotor speed; s = slip: f_1 =

primary supply frequency; p = number of pole pairs; W_r = effective number of turns in series per pole of the regulating winding R; W_s = effective number of turns per phase of the secondary stator winding S; e_r = induced e.m.f. in a single turn of the regulator winding; E_ϕ = e.m.f. induced in the turns of the regulator winding included in the pole pitch (diametral e.m.f.); E_r = regulating voltage; E_2 = secondary stator winding voltage; 2α = angle between a pair of brushes of the same phase (symmetrical displacement).

The rotating field cuts the conductors of the regulator winding R, which is fixed relatively to the primary winding P, at a velocity which is always constant, regardless of the speed of the rotor, but the absolute velocity with regard to the stator depends on the rotor speed and, therefore, on the slip. As the rotor revolves in the opposite direction to that of the rotating field, when the machine is stopped the speed of this rotating field relatively to the stator will still be n_c, and it will be equal to zero at synchronous speed, while at higher speeds the direction of rotation will be reversed.

The voltage measured on the commutator over the distance of the pole pitch has a constant value and is independent of the rotor speed. The e.m.f. induced in the regulator winding has the frequency f_1 of the supply network, while the e.m.f. measured at the commutator between two brushes of the same phase, as a result of the action of the commutator, is sf_1; it is, therefore, equal to the induced e.m.f. of the rotating field in the stator winding S. The rotor assembly of the Schrage machine, therefore, behaves as a rotary voltage and frequency converter.

If e_r is the e.m.f. induced in a single turn of the regulator winding R, then se_r will be the e.m.f. induced in a single turn of the stator winding S and, therefore,

$$e_r W_r = \pm s e_r W_s \quad .. \quad .. \quad .. \quad (1)$$

so that

$$s = \pm W_r / W_s \quad .. \quad .. \quad .. \quad (2)$$

Slip s being determined by

$$s = (n_c - n)/n_c \quad .. \quad .. \quad .. \quad (3)$$

the mechanical speed n of the rotor can, therefore, be expressed as

$$n = [(W_s \pm W_r)\, n_c]/W_s \quad .. \quad .. \quad (4)$$

The rotor speed n can thus be varied by shifting the brushes connected to the ingoing conductors of the secondary winding S in the opposite direction to that

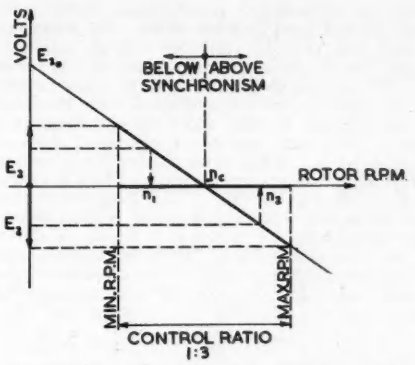


Fig. 4. **Relationship between rotor speed and slip voltage.**

of the other line of brushes connected to the outgoing conductors of the same winding, so that they will include a greater or a smaller number of turns W_r of the regulator winding R. The rotor speed is, therefore, continuously adjustable, and the regulating ratio n_{max}/n_{min} is only dependent on the ratio W_r/W_s.

The speed variation was more easily appreciated by considering the number of turns involved, but the correct explanation is based upon consideration of the voltages. Neglecting the slight phase difference between E_2 and E_r, we have

$$sE_2 \pm E_r = R_2 I_2,$$

and as slip is defined by eq. (3), we obtain

$$n = E_2 \pm (E_r + I_2 R_2)\, n_c / E_2 \quad .. \quad (5)$$

Under no-load conditions, the drop $R_2 I_2$ can be disregarded, and eq. (5) then becomes

$$n = (E_2 \pm E_r)\, n_c / E_2 \quad .. \quad .. \quad (6)$$

which is similar to eq. (4).

The turns of the regulator winding included between the two brushes can vary from W_r to 0, and this means that a wide range of voltages can be taken from the commutator between 0 and the maximum voltage E_ϕ. The speed range at no-load is, therefore, determined by

$$n_{min} = (E_2 - E_\phi)/E_2 \text{ and}$$

$$n_{max} = (E_2 + E_\phi)/E_2$$

and for any position of the brushes, we have

$$E_r = E_\phi \sin \alpha.$$

The voltages E_r are proportional to the various chords connecting the positions of the brushes on the commutator, on the circumference representing the commutator in the ideal case. In the case of a two-pole machine, the maximum value is equal to the diameter.

If the brushes are brought face to face, that is, if the brushes of the same phase are on the same commutator segment, each phase of the secondary winding will be in short-circuit, and the voltage collected at the brushes will be zero, so that the motor will then

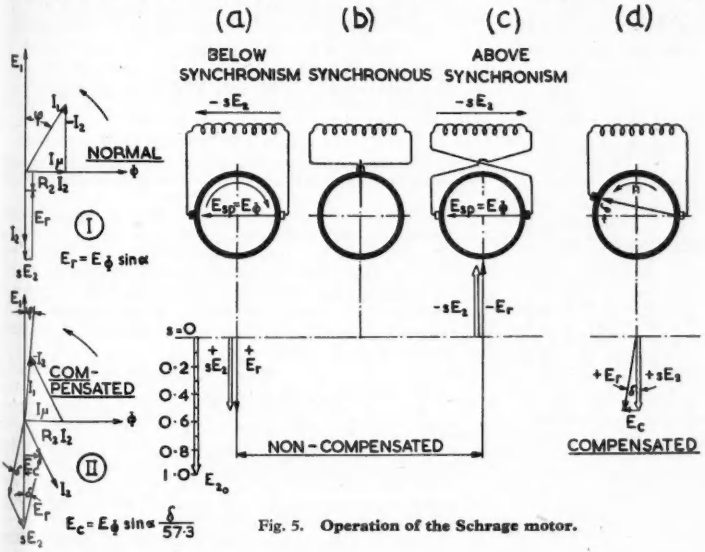


Fig. 5. **Operation of the Schrage motor.**

behave as an ordinary asynchronous motor running at its normal asynchronous speed. By crossing the brushes and rotating until the overlap reaches 180 degrees, the voltage at the commutator will be reversed, and will continue to increase until it finally covers the full diameter on the circle diagram, but its direction will be opposite to that under sub-synchronous conditions, since the machine is operating above synchronous speed. Brush reversal, therefore, reverses the voltage in the secondary circuit, and this is expressed by the $\pm$ sign in eq. (6). The voltage and frequency induced in the secondary winding S depend on the rotor speed, and the variation is linear, as shown in Fig. 4. The functioning of the Schrage motor under no-load is schematically shown for three characteristic brush positions in Figs. 5a, b, c, the voltages being determined from the simplified vector diagram of Fig. 5-I.

2B. POWER FACTOR CONTROL

Phase variation is obtained by simultaneous displacement in the same direction of the same pair of brushes on the commutator. This does not cause a variation in voltage, since the number of turns of the regulator winding included between the two brushes is unchanged. Frequency, moreover, is only dependent on the slip.

In the case of medium or high power motors, running below the synchronous speed, a displacement of only one of the two brushes through an angle δ contrary to the direction of rotation will cause the voltage E_r collected from the commutator to be phase-displaced with regard to the direction of the secondary slip voltage sE_2. Consequently, the secondary current I_2 will be leading the voltage, and will give rise to a resultant $R_2 I_2$ with two components, one of which is in phase with sE_2 and causes a slight drop in speed, while the other, perpendicular to this, is known as the compensating voltage E_c. This provides complete or partial compensation of the magnetizing current $I\mu$ and reduces the phase difference φ between I_1 and E_1; it, therefore, improves the power factor of the motor, as shown in diagram II of Fig. 5.

The value of the compensating voltage E_c depends on the angular displacement δ:

$$E_c = E_\phi \sin \alpha \frac{\delta}{57{\cdot}3} \quad \ldots \quad \ldots \quad \ldots \quad \ldots \quad (7)$$

where δ is in degrees, and $E_\phi \sin \alpha = E_r$.

At synchronous speed, no voltage is collected at the commutator, and, therefore, it is not possible to obtain a self-compensating effect. In brief, it can be said that the motor is non-compensated when the two lines of brushes are displaced through 180 electrical degrees in opposite directions, while for the compensated motor, one line of brushes is shifted through the same given angle and the other through a slightly smaller angle; in the former case the system will, therefore, have to rotate at slightly greater speed than in the latter.

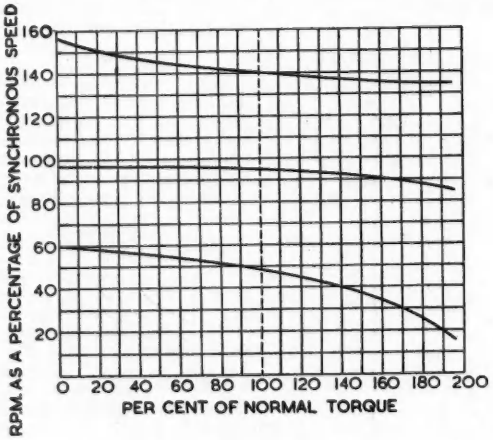


Fig. 6. Velocity as a function of torque for a medium power motor with a regulation ratio of 1 : 3.

3. OPERATING CHARACTERISTICS

3. (a). *Starting.*

The motor is started up with the brushes in their minimum-speed position, an auxiliary contact being provided so that the main switch can only be closed with the brushes in this position. The starting is normally achieved without any resistance in the secondary, since the motor torque is amply sufficient. For gradual starting, however, auxiliary resistances can be used in the secondary circuit, as in the case of an asynchronous motor, or more economically in the primary circuit if there are more than three phases in the secondary.

3. (b). *Motor Speed as a Function of Torque.*

For a medium power motor, with a regulation ratio of 1 to 3, the variation of speed against normal torque is given in Fig. 6. The lowest curve is for the brushes adjusted to low speeds for starting, and the uppermost curve is for maximum speed with the brushes shifted to their extreme positions. Intermediate curves can be obtained by shifting to intermediate brush settings, and it will be seen that at higher speeds, the speed variation with increasing torque is not very considerable. The drop of the lowest curve is due to the losses in the secondary circuit which includes the regulator winding.

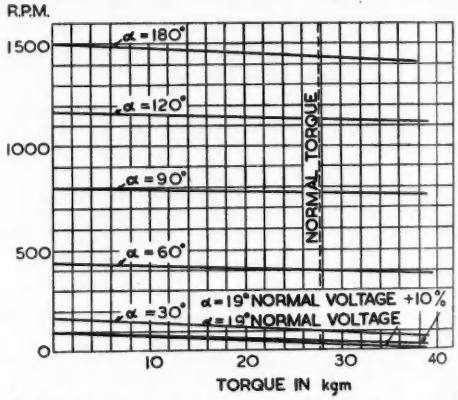


Fig. 7. Motor speed as a function of torque for a special motor with a regulating ratio of 1 : 50, for various positions of the brushes.

Similar curves are given in Fig. 7 for a special motor with an exceptional regulating ratio of 1 : 50 at full load, with its power varying from 40 to 0·8 kW between 1420 and 29 rpm. Owing to the unusually low value of the minimum speed, the lowest curve is considerably affected by slip, and an increase of 10 per cent in the supply voltage will appreciably decrease the slope of this curve.

3. (c). *Efficiency and Power Torque as Functions of Torque and Speed.*

Even at low speeds, the efficiency and the power factor remain favourable. Fig. 8 gives three curves representing efficiency for the three characteristic brush positions (maximum, mean and minimum speeds) for a six-pole motor of 27/9 kW, 1410/470 rpm, with synchronous speed at 940 rpm. For normal torque values, efficiency decreases only slightly at super-synchronous speeds, the decrease being more marked

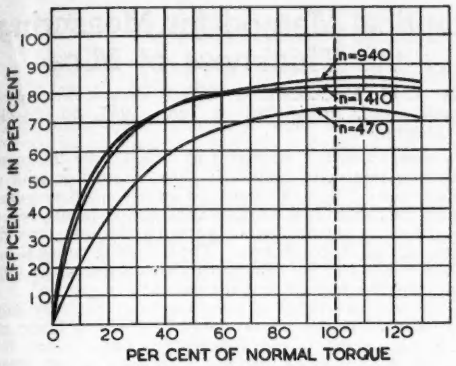


Fig. 8. **Efficiency as a function of torque.**

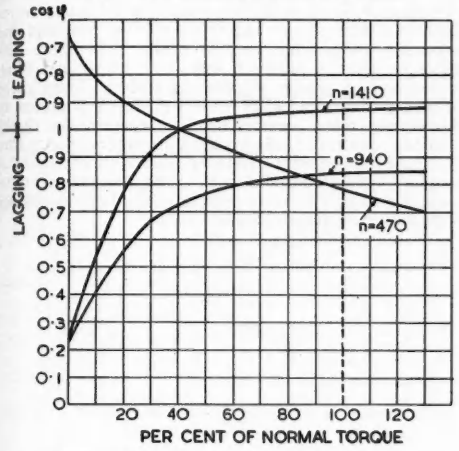


Fig. 9. **Power factor as a function of torque.**

at lower speeds. The conditions are in any case much more favourable than for a normal asynchronous motor running at the lower speeds under consideration.

Fig. 9 gives the corresponding three curves for the power factor. In the case under consideration, the power factor is very good at all speeds and loads up to one-third of the normal torque. For a constant torque, the variation of the power factor with speed can be determined from curves similar to those of Figs. 8 and 9. The results are summarized in the following table, assuming constant torque (synchronous speed = 940 rpm):

rpm	efficiency η	power factor cos φ
1410	81 per cent	0·95 leading.
940	84 per cent	0·84 lagging.
470	74 per cent	0·78 lagging.

3. (d). *Comparison of the Shunt-Commutator Motor with other Regulating Systems.*

(i) The normal asynchronous motor, with resistances inserted in the rotor circuit, is a primitive and unstable arrangement which does not give continuous regulation; its efficiency is extremely low, and it is only applicable in cases where regulation does not have to be continuous.

(ii) The Ward-Leonard group: This is a rational system, and the only one that can be compared with the shunt-commutator motor. However, besides being a more complicated design, embodying three machines apart from the control motor, which is generally an asynchronous machine, its efficiency is less favourable. However, it can be used to good advantage in special cases where ample speed regulation is required at high power ratings.

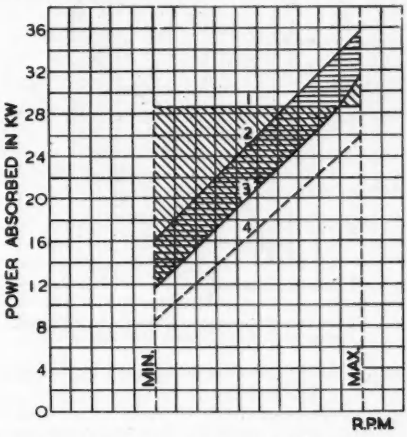


Fig. 10. **Comparison of power absorbed by various regulating systems, for a regulating ratio of 1 : 3.**

As an example, Fig. 10 gives comparative curves for the power absorbed at the same maximum power output of 26 kW represented by the straight line 4, for a regulation with constant torque and a ratio of 1: 3. Curve 3 shows the power absorbed by the Schrage motor, and curve 2 that of a Ward-Leonard group, while curve 1 is for the normal asynchronous motor, the shaded areas representing the differences in absorbed power. In the example considered, the asynchronous motor has lower losses than the Schrage motor over the range 1,500-1,400 rpm., and lower losses than the Ward-Leonard group over the range 1,500-1,150 rpm. The comparison shows that the shunt-commutator motor is more suitable because of its lower energy consumption; furthermore, it is less costly and simpler in design than the Ward-Leonard group with normal asynchronous motor control. Comparing the two systems, it has been calculated that the shunt-commutator motor is approximately 30 per cent cheaper for 30/10 kW and 1,400/470 rpm, and costs about the same for 150/50 kW and 600/200 rpm. To be complete, it should be added that at higher powers the commutator motor yields its position to the normal asynchronous motor regulated by a commutator motor, for instance, a Brown-Boveri-Scherbius type of machine, inserted in its rotor circuit. This more elaborate system does not permit very large speed variations to be obtained, but it is in current use with asynchronous motors for the control of medium and large-size equipment.

(iii) Characteristic curves as functions of motor speed and load: The influence of speed regulation obtained at constant torque by varying the angle α between the two brushes is shown in Fig. 11, applicable to the case of a motor of 65/7·2 kW and 1,170/130 rpm. The α-curve has a sinusoidal shape, and also gives a qualitative indication of the e.m.f. collected at the commutator between the two brushes. The primary current I_1, except for variations of efficiency and power factor, is a parameter of the power delivered and decreases with speed. Furthermore, the secondary current I_2 is a torque parameter; since torque remains constant during the speed variation, I_2 is also practically

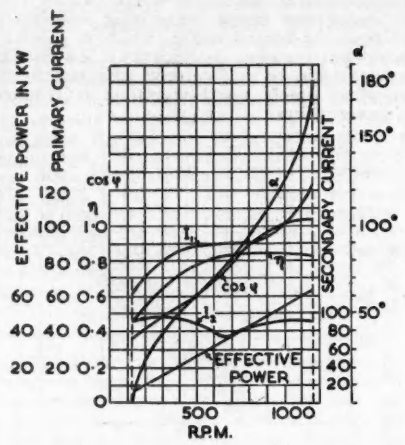


Fig. 11. **Characteristic curves as functions of speed for a motor of 65/7·2 kW and 1170/130 rpm.**

constant. The torque C is determined by the equation

$$C = k I_2 \phi \quad .. \quad .. \quad .. \quad (8)$$

where k = proportionality constant, I_2 = secondary rotor current, nearly constant, ϕ = flux of the machine, also practically constant. With I_2 maintained at a constant value during speed variation, the stator of the motor can be protected by means of simple thermal bimetal relays.

3. (e). *Braking and Torque as Functions of Speed.*

The shunt-commutator motor, like the asynchronous motor, can function as a generator and feed back energy into the grid if its speed is suddenly reduced or if it is externally driven, as, for instance, when it is coupled to hoisting equipment which runs freely when the weight is let down again. For emergency stopping, the motor can be brought to a standstill from maximum speed in $1^1/_2$ seconds by braking against the current and resistances of the secondary. The braking can be made even more effective by simultaneously increasing the supply voltage by means of an auto trans-

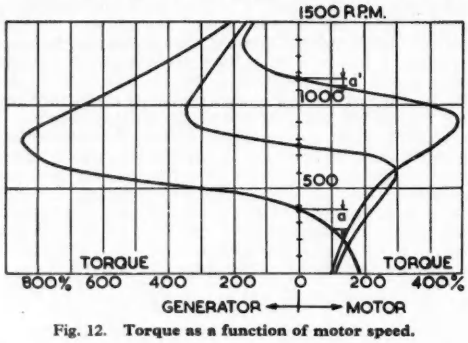


Fig. 12. **Torque as a function of motor speed.**

former. The variation of torque against rpm is shown in Fig. 12. The heavy curve is applicable where the brushes are close to each other; the lower left-hand curve applies where the brushes are at minimum-speed setting, for starting, and the upper right-hand curve refers to brushes in the maximum-speed position. These curves again show that the speed drop "a" is slightly greater at low speeds, less pronounced at maximum speeds, and normal at synchronous speed.

(To be continued.)

Optical Method for Measuring the Thickness of Mica

By N. M. MELANCHOLIN. (From *Vestnik Electropromyshlennosti, Russia*, No. 6, June, 1947, pp. 23-24, 1 illustration.)

IN some cases it is necessary to measure the thickness of mica leaf more accurately than is possible with a micrometer (0·01-0·005 mm). The method of measurement described in this article is based on the difference of colouring when mica is observed in the light through two crossed Nicol Prisms. The colouring depends on the difference of the light paths which is proportional to the double refraction and the thickness of the leaf.

Of the two kinds of mica used commercially only Muscovite, which has a double refraction figure of 0·005, is suitable to give sufficiently accurate results for this method. Some thicknesses of mica do not show very distinct colouring, such as the leaves used for the manufacture of micanite, for which the thickness varies between $0{\cdot}02 \pm 0{\cdot}005$ and $0{\cdot}03 \pm 0{\cdot}005$ mm and the colouring varies from grey to white. The more distinct colours, i.e. yellow, orange, red, purple and blue, correspond to greater thicknesses and to obtain these colours a thin mica leaf of known thickness is added to the sample being measured. To determine its thickness, the colouring of the mica leaf is compared with that of a calibrated quartz wedge or with a set of leaves of standard calibrated thicknesses. The measuring instrument shown in Fig. 1 has been developed for measuring mica leaves in industrial laboratories.

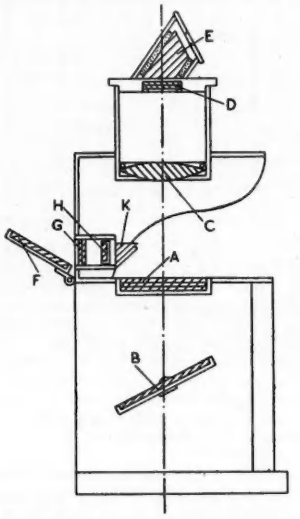


Fig. 1

A—Polarizer with calibrated mica leaf added; B—mirror for lighting polarizer; C—lens; D—nicol prism at right angle to A; E—prism, which permits observation in a horizontal direction which is less tiring than vertical observation; F—quartz wedge; G—mirror for lighting F; H—polarizer; K—prism directing the light which has passed through F to the lens C.

The quartz wedge is divided by thin black lines into four sections equivalent to thicknesses of less than 15μ, 15-25μ, 25-35μ and more than 35μ. The sample is placed on top of A and turned by hand until its colouring becomes darkest. This colouring is now compared with the corresponding zone of the wedge, which can be viewed simultaneously, and the thickness of the leaf is thus established.

The results obtained with this method are accurate to within 2-3μ.

A New Lacquered-Wire Testing Apparatus

By R. FRIZA. (From *Elektrotechnik und Maschinenbau*, Vol. 65, No. 1/2, January/February, 1948, pp. 14-17, 4 illustrations.)

IN the manufacture of lacquer-covered wires, insulation defects cannot always be avoided, but they should not exceed certain limits. The wire is therefore subjected to random tests in a lacquered-wire testing apparatus,* which determines the number of defects. The test length and the permissible number of faults are specified for the various types of wires and wire diameters ; for instance, if the test length is 15 metres, the number of faults should not exceed 15.

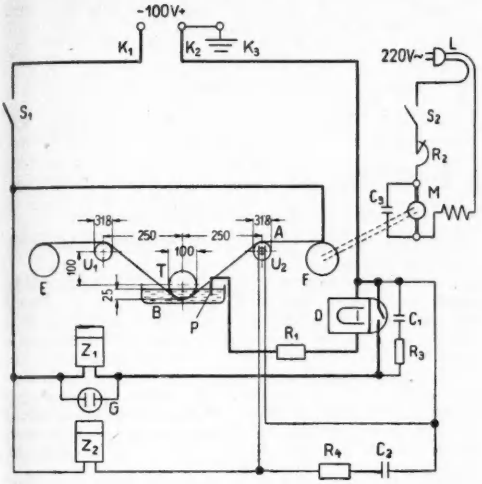


Fig. 1. **Diagrammatic arrangement of the apparatus.**
A, wire being tested; B, electrolytic bath; D, moving coil relay; E, unwinding device; F, winding device; G, pilot lamp; K_1, K_2, terminals for the test voltage; M, driving motor; P, electrode; R_2, motor regulating rheostat; T, dipping wheel; U_1, $_2$, guide pulleys; Z_1, fault counter; Z_2, wire-length counter.

In the new apparatus, see Fig. 1, the coil is fixed on the rolling device *E*. The wire *A* runs at a speed of 20 cm/sec between the guide pulleys U_1 and U_2, passes under a dipping wheel *T* through a bath *B* with a 0·5 per cent solution of cooking salt, and is then wound on the winding device *F* which is driven by an electric motor *M* through a worm wheel drive. The beginning of the wire is insulated and connected via a terminal flange on the winding device to the terminal K_1 which is linked up to the negative pole of a d.c. voltage of 100 volts. The electrode *P* in the bath is connected through a moving coil relay *D* and a series resistance R_1 to the terminal K_2 on the earthed positive pole of the d.c. voltage. Whenever a defective portion of the wire passes through the bath, a current will flow through the relay *D* which then closes another circuit so that a glow-lamp will light up and the fault-counter Z_1 will switch over to the next number.

The length of wire passed is indicated by the length counter Z_2 which receives a current impulse from the same source of energy through a contact on the guide U_2 for each revolution of the pulley. The pulley has a circumference of exactly 10 cm, so that the insulation of the wire can be accurately checked every 10 centimetres.

The motor *M* driving the winding device is fed from the a.c. mains through a cable *L*, and controlled by a rheostat R_2. With this rheostat the motor is slowly brought up to its normal running speed of 20 cm/sec avoiding excessive acceleration during starting which might cause breakage of the wire.

Two switches S_1 and S_2 connected to the rheostat R_2 cut out both voltages when the rheostat is brought back to its stop. The capacitors and resistors $C_{1,2,3}$ and $R_{3,4}$ serve to quench the sparks at the contacts and to eliminate radio disturbance. The resistor R_1 in series with the relay *D* gives a total relay resistance of 10,000 ohms.

The arrangement counts a maximum of 10 current impulses per second and shows faults in the lacquered wire whenever the resistance is equal to 10,000 ohms or less. The circuit does not respond to defects exceeding 11,000 ohms.

Weights are used to check the tension of the wire which has to be adjusted for each wire diameter to the required value, and the load is generated by a brake on the unwinding device.

To operate the apparatus, an a.c. connection of 220 volts and a d.c. voltage source are required. The d.c. source must be capable of supplying a current of up to 30 milliamperes while maintaining the voltage at 100 volts with a constancy of 1 per cent, its positive pole being earthed.

The bearing friction of the guide pulleys and the dipping pulley was investigated, and it was found that for wires having diameters between 0·03 and 0·8 mm, plain bearings were sufficient but ball bearings were used so as to have minimum friction with wires of larger diameters. Tests on the completed assembly showed that, to overcome the friction due to the three pulleys, a weight of 0·44 gram, that is, a small fraction of the tension applied to the wire, is sufficient ; furthermore, the bearings can withstand without damage a tension of 500 grams which is required for wires 0·8 mm in diameter.

It was more difficult than anticipated to get the apparatus to respond to insulation defects of 10,000 ohms or less without being sensitive to merely weak points in the insulation with a resistance exceeding

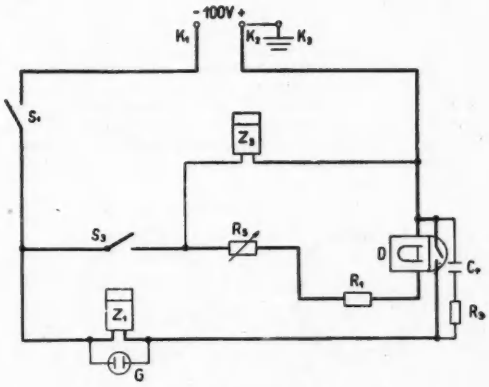


Fig. 2. **Circuit connections for testing the moving coil type relay.**

*The lacquer testing apparatus is generally a mercury apparatus or an apparatus with an electrolytic bath of cooking salt, of the types described in the German Standard Specification DIN. E45453 (see also C. Straubel: New apparatus for testing the insulating layer of fine enamelled wires, Siemens-Zeitschrift, 1921, No. 7/8; W. Hetzel, Measurement of the electrical properties of lacquered wires, E.T.Z., Vol. 64, 1943, pp. 529-532, with bibliography).

11,000 ohms. The currents passing through the relay and its series resistance, with a d.c. source of 100 volts, were 4·76 mA for an insulation resistance of 11,000 ohms, 5·0 mA for a fault of 10,000 ohms, and 10 mA for a fault of 0 ohms. It was found impossible, with the ordinary types of relays available, to obtain a reliable response at 5 mA and then an equally reliable lack of response at 4·76 mA, i.e. a 5 per cent discrimination, particularly when a current of 10 mA had premagnetized the relay. These difficulties were overcome by using a moving coil type relay with impact-free contacts.

The relay was tested in a specially developed circuit (see Fig. 2). A contact cam S_3 was inserted in the negative side of the circuit and driven by a synchronous motor at 50 cps mains frequency, giving a contact of 50 milliseconds and a break of equal duration 50 times per second. This contact acted on the moving coil relay and also on another impulse counter, through a plug-in type resistance box R_5 with which the fault resistance of the test wire was simulated.

The resistor R_5 was set to various values and about 1,000 impulses were produced with the mechanically-driven contact S_3 for each of these resistor settings. The actual number of impulses was registered by the counter Z_3, while the counter Z_1 served to show how many of these actuated the moving coil relay and switched it over to the resistor R_5. The tests showed a response better than 99·9 per cent for resistances of up to 10,400 ohms, and a drop in the response to 0·1 per cent at 10,600 ohms. The moving coil relay was therefore perfectly capable of distinguishing between real insulation faults and mere weaknesses in the insulation, and its transition range was only 200 ohms instead of the 1000 ohms required.

FRANCE

The Floating Bearing

(From *Le Génie Civil*, Vol. 125, No. 13, July 1, 1948, pp. 251-253, 7 illustrations.)

In a note published in *Comptes Rendus*, July 16, 1947, M. Paul Gérard describes a new type of bearing designed to eliminate vibration by allowing the shaft to rotate about its axis of inertia. The shaft makes no contact with the bearing, being supported entirely by fluid pressure. This condition can be maintained even when the shaft is stationary, if the fluid is pressurized by an independent pump.

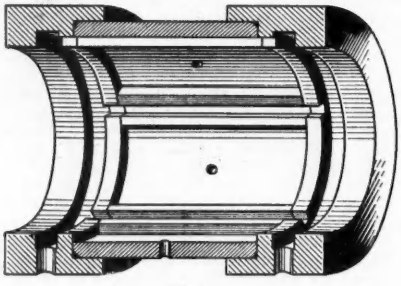


Fig. 1

The principle of operation is very simple. The interior of the bearing has recesses and grooves, as shown in Fig. 1. The recesses are fed with the fluid, under pressure, and the fluid escapes along the grooves, as in Fig. 2. If the shaft is displaced towards one recess, the area through which the fluid can escape from that recess will be reduced and the pressure in it will increase. Conversely, the pressure in the opposite recesses will decrease. Thus, any displacement produces a change in the fluid pressure distribution which returns the shaft to the central position, and, therefore, the equilibrium is stable.

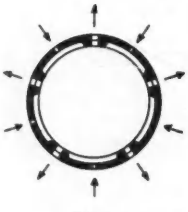


Fig. 2.

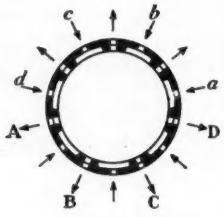


Fig. 3

Fig. 2 illustrates a bearing for a shaft subjected to loads varying in magnitude and direction. In Fig. 3 an arrangement is shown for a shaft under constant load. The lower recesses are fed with fluid under pressure, and each of the lower grooves, A, B, C and D, communicates with the opposite recess by the orifices a, b, c and d, respectively. The fluid finally escapes along the upper grooves. If the shaft is displaced towards an upper recess, the pressure there will increase, both because the area of escape is reduced and because the area of supply (the lower groove) is increased.

Fig. 4 shows a spindle grinder fitted with two bearings of the type shown in Fig. 2. These bearings were fed with ordinary water at 70 psi. They were run for 50 hours without wear, at speeds in the region of 18,000 rpm. The shaft diameter was $1\frac{3}{16}$-in. There was considerable play (0·001-in) between the shaft and the bearings, and the overhang ($4\frac{1}{4}$-in) was intentionally exaggerated. The surface finish obtainable with this grinder was perfectly satisfactory.

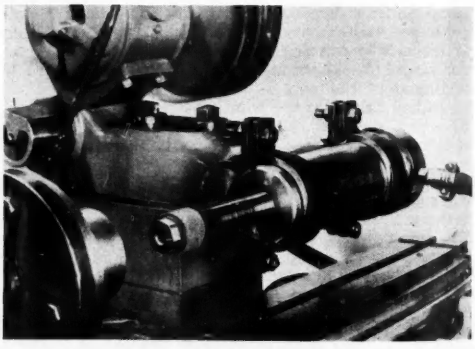


Fig. 4. Surface grinder fitted with two floating bearings.

Three other sets of floating bearings have been successfully run, and an experimental model has been run with compressed air as the working medium. In general, the expenditure of energy in compressing the fluid is justified by the reduction in frictional loss.

Strength Problems Relating to the Design of Centrifugal Pumps

By J. J. HOLBA. (From *Maschinenbau und Wärmewirtschaft*, Vol. 3, No. 1/3, January/March, 1948, pp. 35-40, 5 illustrations.)

GENERAL

THE stress condition of a machine part is defined by the magnitude of the three principal stresses σ_1, σ_2, and σ_3, and various strength theories have been advanced for predicting the condition of failure under any kind of combined stress. The stress σ_G under which failure occurs is a reference stress, subject to definition; in the case of ductile materials, the yield point is taken as the basis for determining working stresses and σ_G can be found by using the "maximum strain energy" theory, viz:—

$$\sigma_G^2 = \tfrac{1}{2}[(\sigma_1 - \sigma_2)^2 + (\sigma_1 - \sigma_3)^2 + (\sigma_2 - \sigma_3)^2] \quad (1)$$

For brittle materials, however, the "maximum strain" theory should be used, viz:—

$$\sigma_G = \sigma_1 - \frac{1}{m}(\sigma_2 + \sigma_3) \quad .. \quad .. \quad (2)$$

where $1/m = \mu = 0{\cdot}3$ = Poisson's ratio.

The working stress σ_v can then be obtained by dividing σ_G by a safety factor S, the value of which depends upon the type of loading. In the following it is proposed to examine the stresses and strains in the casings of single-stage centrifugal pumps, and in the ring sections and tie bolts of multi-stage centrifugal pumps. Let it be assumed here that the ring sections are represented by tubes of constant wall thickness, without any stiffeners, and that stresses and deformations in all planes perpendicular to the pump axis are equal.

STRESS CONDITIONS IN A CYLINDRICAL CASING DUE TO INTERNAL PRESSURE AND AXIAL FORCES

The stresses in a tube (casing) due to the internal pressure p (kg/cm²) are as follows:—

(a) *The tangential stress*

$$\sigma_1 = \sigma_t = p\,\frac{r_i^2}{r_a^2 - r_i^2}\left(1 + \frac{r_a^2}{r^2}\right)$$

The maximum value of this stress is, at the inner surface,

$$\sigma_{1\,\max} = \sigma_{t\,\max} = p\,\frac{a^2 + 1}{a^2 - 1} \quad .. \quad .. \quad (a)$$

where $a = r_a/r_i$. The tangential stress is a tensile stress as shown in Fig. 1.

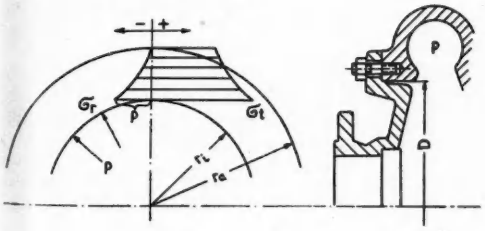


Fig. 1

Fig. 2

Fig. 1. **Tangential and radial stresses in a tube of circular cross-section due to internal pressure.**

Fig. 2. **Bolted joint of a single-stage volute casing pump.**

(b) *The radial stress* can generally be represented by

$$\sigma_r = p\,\frac{r_i^2}{r_a^2 - r_i^2}\left[1 - \left(\frac{r_a}{r}\right)^2\right]$$

The radial stress, however, is a compression stress, the maximum value of which can be obtained, for $r = r_i$, when

$$\sigma_{2\,\max} = \sigma_{r\,\max} = -p \quad .. \quad .. \quad (b)$$

(c) *Stresses due to axial forces.* If it is assumed that the axial forces are uniformly distributed over the cross-section, then

$$\sigma_3 = \sigma_z = \frac{\kappa p}{1 - a^2} \quad .. \quad .. \quad (c)$$

The sign of the coefficient κ depends on the type of cover, and indicates whether the stress is tensile or compressive. If the tube is closed at each end by covers, which are held together by tie bolts as in the case of ring-section type multi-stage pumps, these tie bolts will be pre-stressed to such an extent that, when the pump is in operation, no leakage can occur due to partial unloading of joint faces effected by the internal pressure set up. When running, the internal pressure reduces the initial compression stress in the tube (ring sections) giving a value of $\kappa = 1{\cdot}2$ to $2{\cdot}5$. The lower value relates to the case where a jointing material is used for making the casing leak-proof, and the higher value applies to metal-to-metal joints. For the closed tube $\kappa = -1$, whereas for the open tube $\kappa = 0$ and, therefore, the stress σ_z also becomes zero.

For ductile materials, such as steel or cast steel, we obtain, by introducing eqs. (a), (b), and (c) into eq. (1).

$$\sigma_v = \frac{p}{a^2 - 1}[3a^4 + (\kappa + 1)^2]^{\frac{1}{2}} \leqslant \sigma_{\text{permissible}} \quad .. \quad (3)$$

and from this

$$a^2 = \frac{\left(\frac{\sigma_v}{p}\right)^2 + \sqrt{3\left(\frac{\sigma_v}{p}\right)^2 + \left[\left(\frac{\sigma_v}{p}\right)^2 - 3\right](1+\kappa)^2}}{\left(\frac{\sigma_v}{p}\right)^2 - 3} \quad (4)$$

It is evident from eq. (4) that with $(\sigma_v/p)^2$ approaching the value of 3, a tends towards infinity Allowing a maximum error of 1·5 per cent for the wall thickness and with $\sigma_v/p \geqslant 10$ in eq. (4), the digit 3 can be neglected in comparison with $(\sigma_v/p)^2$.

Under this assumption an approximation for eq. (4) is obtained, viz:—

$$a^2 = 1 + \frac{p}{\sigma_v}\sqrt{3 + (1+\kappa)^2} \quad .. \quad (4a)$$

In the special case of the open tube with $\sigma_z = \kappa = 0$, eq. (4) becomes

$$a^2 = \frac{\left(\frac{\sigma_v}{p}\right)^2 + \sqrt{4\left(\frac{\sigma_v}{p}\right)^2 - 3}}{\left(\frac{\sigma_v}{p}\right)^2 - 3} \quad .. \quad (5)$$

Neglecting as a first approximation the digit 3, as we have done before, we obtain

$$a = \sqrt{1 + 2\frac{p}{\sigma_v}} \quad .. \quad .. \quad (5a)$$

Expanding eq. (5a) to a binomial series, and neglecting terms of higher degrees than the second, we obtain

$$a \approx 1 + p/\sigma_v, \text{ and with } r_a - r_i = s$$

$$s = \frac{p\,D}{2\,\sigma_v}$$

the well known relation for determining the wall thickness of (thin-walled) tubes. The term " thin-walled " is inherent in the assumption of $\sigma_v/p \geq 10$.

For brittle materials, such as cast iron, we introduce eqs. (a), (b), and (c), into eq. (2), thus obtaining

$$a^2 = \frac{\frac{\sigma_v}{p} + \left(1 + \frac{\kappa}{m} - \frac{1}{m}\right)}{\frac{\sigma_v}{p} - \left(1 + \frac{1}{m}\right)} \quad .. \quad .. \quad (6)$$

As a special case, for the open tube with $\sigma_s = \kappa = 0$ and $1/m = 0{\cdot}3$ we have

$$a^2 = \frac{\sigma_v/p + 0{\cdot}7}{\sigma_v/p - 1{\cdot}3} \quad .. \quad .. \quad (7)$$

AXIAL LOADING OF CASING AND TIE BOLTS DUE TO PRE-STRESSING AND INTERNAL PRESSURE

For cold liquids, tightness of adjoining ring sections is obtained by the insertion of joints subjected to the pressure exerted by the tie bolts. For hot liquids, ground metal-to-metal joints are made. The degree of pre-stressing of the tie bolts depends upon the delivery pressure and will be such that, under operating conditions, the gasket pressure is not completely counteracted by the effect of internal pressure and is still sufficient to ensure tightness of the joints. The additional load which the tie bolts have to sustain in operation depends upon pump design and dimensions.

In the following investigation the effect of the elasticity of the joint material as well as the deformation of covers and flanges are neglected. A joint such as can be found on the suction side of a single-stage volute casing pump is shown in Fig. 2. The force V due to pre-stressing is taken to be a multiple of the force P due to the operating pressure.

$$V = \chi\, D^2 \frac{\pi}{4} p = \chi\, P$$

When the pump is in operation, the force available for tightening the joints is

$$P_d = V - P = (\chi - 1)\,\frac{D^2\,\pi}{4}\,p = \kappa\, P \qquad \text{(d)}$$

Thus, for tight joints it is necessary that $\chi > 1$. Normally, $\chi \geq 2$; for hot water pumps, $\chi \geq 3$.

Multi-stage pumps consist of a number of ring sections which are held together by tie bolts. The simplest instance is that of a tubular casing closed by covers as shown in Fig. 3. The force due to pre-stressing of the bolts compresses the casing, which, therefore, is shortened by the amount

$$\delta_G = l\,\frac{V}{E_G\,F_G}$$

whilst the elongation of the bolts is

$$\delta_s = l\,\frac{V}{E_s\,F_s}$$

The whole system shown in Fig. 3 can be represented by springs, the casing being represented by compression, and the tie bolts by tension springs, as illustrated by Fig. 4.

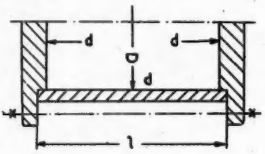


Fig. 3. **Tubular casing closed by covers and held together by tie bolts.**

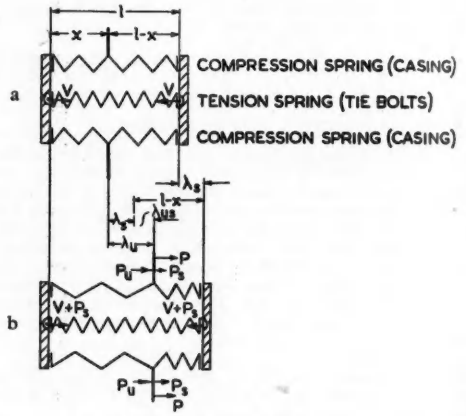


Fig. 4. **The type of loading shown in Fig. 3, represented by a system of springs.**

(a) The system is subjected to pre-stressing only.
(b) The system is subjected to pre-stressing as well as to the operating force.

The total force P is the resultant of the force P_u acting on the casing and the force P_s acting on the tie bolts. The magnitudes of these forces are found from the condition that, when the pump is in operation, the elongation of the casing due to reduced loading equals the elongation of the bolts.

$$P = P_u + P_s \quad .. \quad .. \quad \text{(f)}$$

$$\lambda_G = \lambda_s \quad .. \quad .. \quad .. \quad \text{(g)}$$

$$\text{or} \quad \frac{P_u\,l}{E_G\,F_G} = \frac{P_s\,l}{E_s\,F_s}$$

This equation yields

$$P_s = P\,\frac{1}{1 + \frac{E_G\,F_G}{E_s\,F_s}} \quad .. \quad .. \quad (8)$$

and from eq. (f), the force tending to unload the casing

$$P_u = P - P_s = P\,\frac{1}{1 + \frac{E_s\,F_s}{E_G\,F_G}} \quad .. \quad (8a)$$

In the case of multi-stage centrifugal pumps, the maximum (delivery) pressure is confined to the last stage, whilst in every preceding stage the pressure is less by equal amounts. The deformations due to this type of loading will be considered in the following. Reverting to the representation of forces by springs as illustrated by Fig. 4, we can now imagine the force P which is set up by the operating pressure, at a distance x, in which case, according to Fig. 4b, its component forces will be P_u and P_s.

$$P = P_u + P_s \quad .. \quad .. \quad (h)$$

The component P_u reduces the loading of the compression spring over the length x and extends it by an amount λ_u. The remaining length of the compression spring $(l - x)$ is subject to the action of the remaining force P_s; this part of the spring is compressed by the amount λ_{us}. The same force P_s increases the tension spring and elongates it by λ_s. We obtain, therefore,

$$\lambda_u = \lambda_{us} + \lambda_s \quad .. \quad .. \quad .. \quad (9)$$

or, generally, according to Hooke's Law, $\lambda = lP/FE$,

$$\frac{P_u}{E_G F_G} = \frac{P_s}{E_G F_G} + \frac{P_s}{E_s F_s} \quad .. \quad .. \quad (i)$$

where F_G and F_s are the cross-sectional areas of the casing and tie bolts respectively.

Substituting into this equation for P_u from eq. (h), and transforming, we get

$$P_s = P \frac{x}{l} \frac{1}{1 + \dfrac{E_G F_G}{E_s F_s}} \quad .. \quad (10)$$

This relation, first established by Hanffstengel, comprises the following special cases:—

(a) $x = 0$. The force P is applied at the joint face. Then, according to eq. (10), the additional force in the tie bolts is $P_s = 0$, i.e. $P_u = P$. The total operating force relieves the gasket pressure to $V - P = \kappa P$, whereby $\kappa = \chi - 1$. Because $P_s = 0$, no deformation takes place. This case represents the type of loading shown in Fig. 2.

(b) $x = l$. According to eq. (10), the expression for P_s becomes identical with that of eq. (8).

The casing is relieved by $P_u = P - P_s$, and, therefore, $P_u = P/(1 + E_s F_s/E_G F_G)$. The remaining force acting on the gasket is

$$P_d = V - P_u = P[\chi - 1/(1 + E_s F_s/E_G F_G)] = \kappa P.$$

Eq. (10) is generally applied to multi-stage centrifugal pumps. Since each stage produces the pressure p_1, a stepped distribution of the operating force is obtained, as shown in Fig. 5.

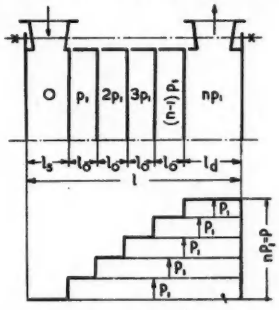


Fig. 5. **Diagrammatic arrangement of the casing of a multi-stage centrifugal pump with n stages, showing the distribution of the operating force.**

The force $P_1 = p_1 D^2 \pi/4$ of the first stage acts over the length $x_1 = l - l_s$.

The force $P_1 = p_1 D^2 \pi/4$ of the second stage acts over the length $x_2 = l - (l_s + l_0)$.

The force $P_1 = p_1 D^2 \pi/4$ of the nth stage acts over the length $x_n = l - [l_s + (n-1) l_0]$.

From the distribution of forces and from the distances over which they act, the additional bolt force P_{s1} of the tie bolts can be calculated with the aid of eq. (10), and with $k = E_G F_G/E_s F_s$.

$$P_{s1} = \frac{P_1}{1+k} \frac{1}{l} (l - l_s)$$

$$P_{s2} = \frac{P_1}{1+k} \frac{1}{l} [l - (l_s + l_0)]$$

$$P_{sn} = \frac{P_1}{1+k} \frac{1}{l} [l - \{l_s + (n-1) l_0\}]$$

$$\sum_1^n P_{sn} = \frac{P_1}{1+k} \frac{1}{l} \left[nl - nl_s - \frac{n}{2} (n-1) l_0 \right] \quad (11)$$

with $nP_1 = P = np_1 F = pF$, eq. (11) can be written

$$P_s = \sum_1^n P_{s1} = \frac{P}{1 + \dfrac{E_G F_G}{E_s F_s}} \left[1 - \left(\frac{l_s}{l} + \frac{n-1}{2} \frac{l_0}{l} \right) \right] \quad (12)$$

which is the additional bolt force due to a stage-by-stage increase of the operating load. The force P_s is smaller than force P relating to the instance where the operating force was constant over the whole length of the casing (Fig. 3). This is of importance inasmuch as, in accordance with eqs. (12) and (h), the force P_u relieving the joints is greater in this case and should be carefully considered in order to prevent leakage.

The maximum relieving force in the last stage is obtained from eqs. (12) and (h).

$$P_u = P \left\{ 1 - \frac{1}{1 + \dfrac{E_G F_G}{E_s F_s}} \left[1 - \left(\frac{l_s}{l} + \frac{n-1}{2} \frac{l_0}{l} \right) \right] \right\} \quad (13)$$

The compressive force P_d the last stage, taking into consideration the force $V = \chi P$ due to pre-stressing and eq. (12),

$$P_d = P \left\{ \chi - \left(1 - \frac{1}{1 + \dfrac{E_G F_G}{E_s F_s}} \left[1 - \left(\frac{l_s}{l} + \frac{n-1}{2} \frac{l_0}{l} \right) \right] \right) \right\} \quad (14)$$

or, substituting κ for the term in the bracket.

$$P_d = \kappa P$$

where $\kappa < \chi$

RUSSIA

Permissible Limits and Measurement of the Roughness of Machined Surfaces

By P. E. DYACHENKO. (From *Stanki i Instrument*, No. 9, 1947, pp. 17-20, 8 illustrations.)

EXPERIMENTS have shown that the measured surface roughness of a machined metal is usually rather different from that which may be expected from calculations based on the shape of the tool used for machining.

When the cutting speed and the rate of longitudinal feed are changed (the two conditions which mainly determine microgeometry) there is a range in which the roughness caused by the cutting edge changes rapidly. Fig. 1 shows H_{max}, the measure of irregularity as a function of the cutting speed v and the feed s. The diagrams in the lower half of the figure show the limits of the various ranges in which the roughness characteristics differ from each other. Calculated formulae for determinating the measure of roughness are also given.

For brittle materials, $H_{max} = f(v)$ is represented by the straight line *A B D E*, but for ductile materials, which are the materials mostly used in machine design, the applicable curve is *A B C E*. The latter can be divided into three sectors: between *A*-*B* and *D*-*E* the microgeometry is independent of the speed v, whereas between *B* and *D* it changes rapidly with v. The peak *B C D* is explained by the character of the plastic deformation in front of the cutting edge. The width of the plastic zone in front of the cutting edge varies according to the law applying to the depth of surface roughness. The maximum depth appears to correspond to a cutting speed of 20 m/min whatever the rate of feed may be.

Within the range of speed enclosed by *B C D*, plastic flow of the metal occurs in front of the cutting edge, and the metal itself fulfils the function of a cutting edge, but as its width is constantly changing, the shape of the cutting edge cannot be accurately determined and $H_{max} = f(v)$ can only be established experimentally. The face of a disk of the metal under examination is turned, and the roughness is measured by means of a compound microscope from the centre to the periphery.

Where the plastic deformation is small—sectors *A B* and *D E* of Fig. 1—the development of roughness differs from that obtained when the plastic deformation is large, and it may vary with the rate of feed. The right half of Fig. 1 shows the curve *FKL* for the relation $H_{max} = f(s)$.

The formula for this curve is

$$H_{max} = \frac{s^2}{8R} + H_y \cdot \left(1 - \frac{s^2}{2R^2}\right),$$

where R is the radius of the front edge.

When using a broad cutting tool, i.e., a large radius R, and when the feed is slow, the relation $H_{max} = f(s)$ will not be the same for a finishing operation as it will be for rough turning.

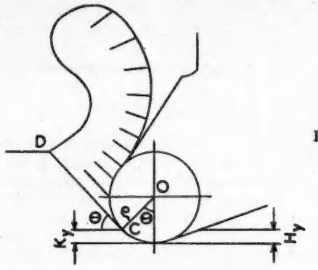


Fig. 2

From Fig. 1 it may be seen that the above formula is applicable to intermediate and rough turning and is represented by curve *KL*. For finishing, however, the relation at first follows the horizontal line *MN* and subsequently, between *N* and *K*, approaches and finally merges into curve *KL*.

The difference between the results obtained experimentally for the surface roughness and those obtained by calculation is explained by elastic deformation of the machined metal during machining. The metal is cut, not at the lowest point of the cutting edge, but at a somewhat higher point. This distance depends on the rounding-off of the cutting edge, which occurs even when the cutting tool is sharpened. In Fig. 2, ρ is the radius of the cutting edge. The line on which the metal is sheared is *CD* and its angle θ. It can be assumed that the point *C*, where the highest stress occurs, is also the point where the chip begins to shear from the metal. The height of point *C* above the lowest point of the cutting tool will be $\rho(1 - \cos\theta)$. The remaining layer of metal is depressed by the movement of the tool, but after the latter has moved on the metal rises again due to its elasticity.

The height of metal restored can be said to be approximately

$$H_y = \rho(1 - \cos\theta)$$

Fig. 3 shows a cutting tool *KFL*. If the metal were of an entirely plastic nature, a cutting tool *MEN* of dimensions reduced by H_y would produce the same

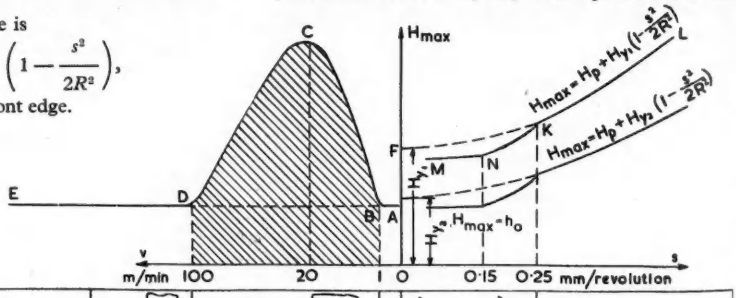
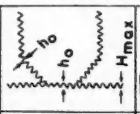
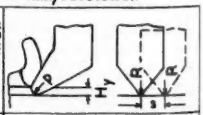


Fig. 1

DIAGRAM OF FORMATION OF ROUGH SURFACE				
CALCULATED FORMULAE	$H_{max} = h_o$ $H_{max} = H_p + H_y\left(1 - \frac{s^2}{2R^2}\right)$	$H_{max} = (A + Bv)e^{Kv} + C$	$H_{max} = h_o$	$H_{max} = \frac{s^2}{8R} + \rho(1 - \cos\theta)\left(1 - \frac{s^2}{2R^2}\right)$
RANGE	SMALL PLC. DEFORMS.	LARGE PLASTIC DEFORMATIONS	FINAL MACHINING	ROUGH AND INTERMEDIATE MACHINING

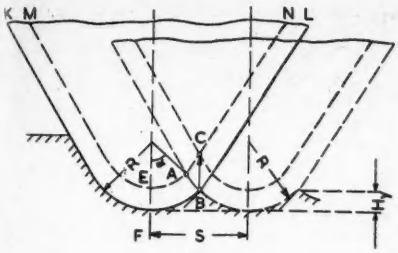


Fig. 3

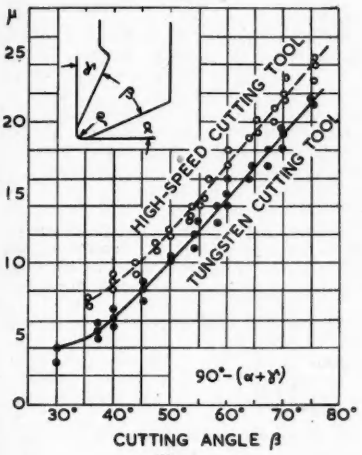


Fig. 4

result, H_y being the height to which the metal rises due to its elasticity.

Fig. 4 shows the angle of the cutting tool for a finishing cut. If the metal had no elasticity, the depth of roughness would be that calculated from the shape of the tool, but because the metal always possesses some degree of elasticity, the metal rises to a different extent at the top and bottom of each depression. Whereas at the top the metal rises by CB, it rises at the bottom only by an amount $AB = EF = H_y$.

The actual depth of roughness can be expressed by the formula

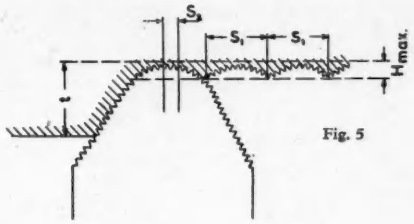


Fig. 5

$$H_{max} = H_p + H_y \left(1 - \frac{s^2}{2R^2}\right)$$

It can be approximated

$$H_y = \rho\,(1 - \cos\theta)$$

and then

$$H_{max} = \frac{s^2}{8R} + \rho\,(1 - \cos\theta)\left(1 - \frac{s^2}{2R^2}\right)$$

Thus, the actual roughness value depends not only on s and R, but also on H_y, which is a measure of elasticity of the metal, for which the radius ρ of the cutting edge and the angle θ at which the chip breaks off must be determined. The radius ρ can only be determined experimentally, usually by obtaining an imprint of the cutting edge in wax or celluloid or by means of a projector or compound microscope. The latter method has been found most convenient, different sizes of circles being superimposed upon the image of the cutting edge under a 500-fold magnification.

The radius can also be obtained from the curves in Fig. 4, which are based on experimental results for different cutting angles with high-speed steel and tungsten steel.

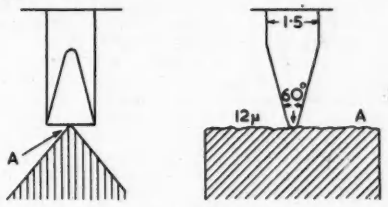


Fig. 6

The value H_y for the height of elastic rise is also obtained experimentally. The sample is machined, the degree of roughness measured and the point entered on the diagram $H_{max} = f(s)$. A curve expressed by the formula

$$H_{max} = \frac{s^2}{8R} + H_y \left(1 - \frac{s^2}{2R^2}\right)$$

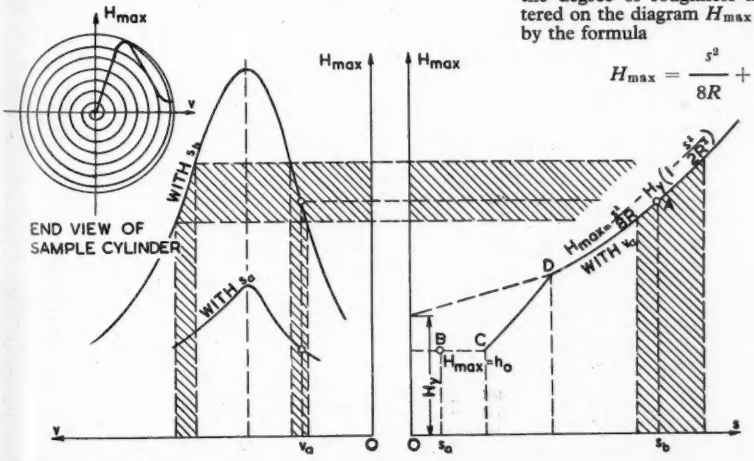


Fig. 7

is drawn through the point. The curve will cut the ordinate at a point F (Fig. 1). The distance OF gives the value for H_y.

H_y can also be calculated by measuring H_{max} and introducing this value into the equation above. If H_y is known, the formula will, either by calculation or graphically, give H_{max} for any given value of the radius R of the front rake.

If the cutting tool is of more than a certain width, the roughness of the surface will be of a different nature.

Fig. 5 shows a cutting tool which has a very slightly irregular surface. With a fast feed s, the roughness caused by the roughness of the tool is of secondary importance and can be neglected. When finishing, however, with a broad tool, and when plastic deformation is small, the roughness is only caused by the roughness of the tool. When the plastic deformation is considerable, the effect of the cutting speed must also be added. For finishing work, it is, therefore, essential to measure the roughness of the cutting edge. The latter is determined either by the maximum depth of the irregularities H_{max} or by the root mean square deflection H_a. H_{max} is measured under a microscope.

The Russian Standard Specifications evaluate the degree of roughness by the r.m.s. value of the depths of an irregular surface and, therefore, it is necessary to do the same with the surface of the cutting tool. This is normally done with an Abbot's Profilometer.

It is suggested using a spatula for the Profilometer measurement in place of the more common needle. The dimensions of the spatula may be seen from Fig. 6. It is of hardened steel instead of the diamond of which the needle is made. The accuracy of measurement is the same in both cases.

Analysis of the problem reveals the required speed of drive and traversing for a permissible degree of roughness.

Fig. 7 shows a graphical method for obtaining these results. H_{max} forms the ordinate of the system, while the abscissae to the left and right give the rate of feed v and the traversing speed respectively. It is necessary to obtain the relation $H_{max} = f(v)$ experimentally for two different traversing speeds, one for a finishing cut with s_a and the other for a roughing cut with s_b. In order to arrive at a curve for $H_{max} = f(v)$, the face of a cylindrical sample is turned and the degree of roughness is measured with a compound microscope from the centre to the periphery. Point A (for s_b and v_a) and point B (for s_a and v_a) are then marked in the system (H_{max}, s). A horizontal straight line is drawn through B and a curve is drawn through A in accordance with the above formula. The straight line terminates in C, $s = 0{\cdot}15 - 0{\cdot}20$ mm/revolution. C and D ($s = 0{\cdot}25$ mm/revolution) are then joined. With these curves drawn, all the required results can be obtained for varying values of s, v and R.

If certain roughness limits are permissible, horizontal dotted lines can be drawn through these. The projections on the abscissa of the sections of the curves between the dotted lines represent the permissible limits for feed and traversing speeds.

Intergranular Corrosion of Cast Austenitic Stainless Steels

By H. A. PRAY. (From *Revue de Métallurgie, France*, Vol. 45, Nos. 1 and 2, January-February, 1948, pp. 19-31, 6 illustrations.)

INTERGRANULAR corrosion of stainless steels has been studied for many years, and certain remedies, such as the addition of small quantities of columbuim or titanium, have been discovered and are being effectively used. Extremely low carbon contents, obtained by the use of oxygen or nickel-oxide, are still in the experimental stage in the production of forged steels, and almost all the literature on this subject refers to worked alloy steels, which are characterized by having a fine grain and a fully austenitic structure. Nothing appears to have been published on systematic research into the susceptibility to intergranular attack of coarse-grained cast stainless steels.

Intergranular corrosion of chrome-nickel austenitic steels in certain media, after heat treatment, is associated with the precipitation of carbides between the primary grains. The great difference in grain size between cast and worked material results in a great difference in the space occupied by the intergranular boundaries. Moreover, forged steels are generally completely austenitic, whereas in cast steels a variable proportion of ferritic constituent is present.

Small experimental ingots, weighing 15 lbs each, were cast in dry sand moulds at 1590 to 1620° C, machined and cut into slabs of about 5-in × 1½-in × ½-in which were heat-treated and then machined into test pieces of 4-in × 2½-in × ¼-in, lapped to give a surface within 20 micro-inches roughness. These steels were of the type 19 per cent Cr, 9 per cent Ni, 0·09 per cent C, 1 per cent Si, 0·80 per cent Mn, 0·03 per cent max. S and P, 0·06 per cent N, but with Cr content varying from 15 to 22 per cent and Ni content varying from 7 to 12 per cent.

Carbides were brought into solution by heating for 30 minutes at 1135° C, and water quenching. The sensitization treatment was carried out for 24 hours at 649° C, followed by air cooling. The best method for testing susceptibility to intergranular corrosion was considered to be that of using boiling 65 per cent nitric acid over 2 to 5 periods of 48 hours, according to speed of attack. Depth of penetration is calculated from loss of weight and increase in electrical resistance by the Kelvin Bridge method.

Results of tests on 30 alloy steels of the 19 per cent Cr, 9 per cent Ni type show that the susceptibility to intergranular attack is due to the precipitation of carbides at the boundaries of the primary grains. The appearance of precipitates in the ferritic phase, which occurred in a certain number of the steels studied, does not produce this susceptibility. The presence of a ferritic zone in the austenitic matrix eliminates this susceptibility, because at precipitation temperatures in the tempered material, carbides are precipitated more readily in the ferrite.

By suitable adjustment of the respective contents of Cr, Ni, N, Si and C in the alloy steels, subjected to tempering, a sufficient amount of ferrite can be produced to decrease considerably, and even eliminate, the tendency to intergranular corrosion which results from subsequent treatment at temperatures at which precipitation of carbides is observed. In alloy steels of this kind, the presence of a ferritic phase is harmful from the forging and rolling point of view, for which reason great care is taken to ensure that the structure of material intended for working is entirely austenitic. However, for use as castings, the proportion of ferrite necessary to reduce or prevent the susceptibility to intergranular corrosion has no harmful effect on the other physical properties, but, on the contrary, seems to be an eminently useful and desirable constituent.

The susceptibility to intergranular corrosion may also arise from incomplete solution of the carbides or from incorrect tempering treatment. The mechanism is as previously described. The minimum tempering temperature necessary for complete dissolution of the carbides greatly depends on the carbon content and varies between 760 and 1260° C or more. The variations in susceptibility caused by incorrect solution treatment are very great and can be controlled by bringing about the formation of a suitable quantity of ferrite.

Drying of Wood with the aid of High Freqency Currents

By A. V. NETUSHIL. (From *Vestnik Inzhenerov i Tehnikov*, No. 4, April, 1947, pp. 141-148, 11 illustrations).

THE main advantage in the drying of wood with the aid of high frequency currents is that the heat is generated inside the wood and is evenly distributed through it. This allows the moisture to escape towards the surface, and reduces the proportion of wood which must be rejected owing to excessive dryness of the surface.

It has been found that the short wave band—up to 50m—produces satisfactory results only with wood samples of small size. For large quantities of wood the efficiency of the generators is too low and the heating is uneven.

Medium waves of 300-1200m have been found most suitable for drying large quantities, the results being more satisfactory qualitatively whilst the process is more economical.

It must be stated, however, that this process is only economical where an ample supply of electricity is available at low rates, as 1 cubic metre of wood requires 250-350 kWh to dry it. For economic reasons, therefore, it should be limited to :—

(1) Partial drying and impregnation of transmission line pylons and wood for port installations.
(2) High quality drying of beams of large section.
(3) Drying and impregnating of valuable wood.
(4) Drying of wood for special applications.
(5) Drying and impregnating of railway sleepers.
(6) Drying of building materials.

While the proportion of rejects with other methods of drying is up to 50 per cent for a 30 per cent final moisture content, and up to 100 per cent for a 22 per cent final moisture content—high frequency drying results in only 5 per cent rejects for a moisture of 12 per cent. The time required for the process is only 2·5 to 3 per cent of that required with other processes.

DESCRIPTION OF THE INSTALLATION.

The process is carried out in a chamber, the walls of which are lined with refractory material, and the wood is placed between the electrodes which are fed from the generator, whilst the chamber is ventilated by a fan driven by a motor of 2-3 kW rating.

The electrodes consist of metal grids arranged horizontally or vertically. With the horizontal arrangement, wood and metal may be placed in alternate layers, or, for thin boards, there can be two or three layers of wood to one electrode.

The vertical arrangement of electrodes is shown in Fig. 1; the distance of the electrodes from the wood is chosen to accord with the required load characteristic of the h.f. generator. When raw wood of circular section is dried, horizontal electrodes are used to cover a certain arc of the circumference the length of which is determined by the kind of wood and the ratio of heart to alburnum.

This method of drying permits the treatment of a certain length, without affecting the remainder which may be required for subsequent impregnation of the section.

IMPREGNATION OF WOOD.

When wood, which has been dried out to a moisture content of 18-25 per cent, is placed, while hot, into a cold watery solution or oil, an intensive absorption of the liquid is observed. The explanation of this is that the residual moisture in the wood exists as steam which condenses, thus creating a vacuum.

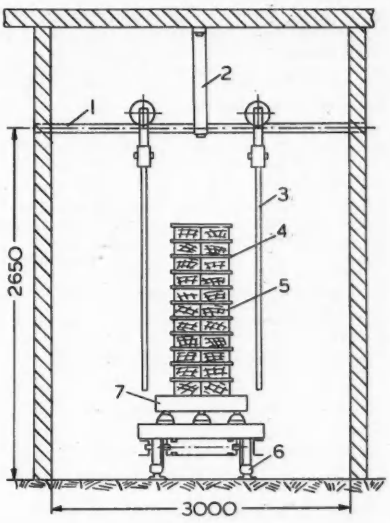


Fig. 1.
(1) guide beam; (2) suspension rod; (3) electrode; (4) spacers (5) beams; (6) narrow gauge rails; (7) trolley.

Impregnating oil completely saturates the alburnum of hot pine; in places where the alburnum is removed, the heart is saturated to a depth of 4-5 mm and the ends of the wood to a depth of 5-6 cm. Watery solutions would saturate the whole of the wood.

When the wood is removed from the impregnating liquid, this is absorbed further from the surface towards the centre, and leaves the surface dry. The latter can therefore be treated to prevent the ingress of water and washing out of the impregnating substance.

The equipment required for impregnation is very simple and consists of:—

(1) Wooden or iron baths.
(2) Containers for impregnating substances.
(3) Pumps for circulating the impregnating substances.

ELECTRIC CHARACTERISTICS OF WOOD.

The intensity of currents generated in the wood depends on the field between the electrodes, the frequency, and the various physical constants which are functions of the properties of the wood. The problem is complicated by the fact that wood is anisotropic and, in addition, inhomogeneous. For the present investigation the characteristics of the wood in the direction perpendicular to the grain is most important.

In practice it is necessary to work with average values of voltage and current density and for these average characteristics are required.

According to Ohm's law the following relation applies for the field potential **E** and current density **J**.

$$\mathbf{E} = z_v \mathbf{J} \text{ or } \mathbf{J} = y_v \mathbf{E} \quad .. \quad .. \quad (1)$$

where z_v and y_v are complex values for the specific resistance and specific conductivity of the wood.

$$z_v = 1/y_v$$
$$y_v = g_v + j\,b_v$$

where b_v is the current leading the voltage by 90 degrees. g_v is the current due to ohmic losses in the wood. These losses can be divided into:—

(a) Ohmic resistance losses which are largely independent of frequency.
(b) absorption losses caused by inhomogeneity of the substance, molecular friction, and high voltage polarization.

$$y = g_v + j b_v = \gamma + \omega\epsilon'' + j\omega\epsilon = \gamma + j\omega(\epsilon - j\epsilon'')$$

where γ is the ohmic loss,
ϵ'' is the absorption loss,
ϵ is the dielectric constant.

$$g_v = \gamma + \omega\epsilon''$$
$$b_v = \omega\epsilon \quad .. \quad .. \quad .. \quad .. \quad (2)$$

In dieletrics where $\gamma = 0$, the complex conductivity is characterized by the complex dielectric coefficient ϵ^* alone, or by ϵ^* and the angle of loss tan δ.

$$\epsilon^* = \epsilon - j\epsilon'' = \epsilon(1 - j\tan\delta)$$
$$y_v = j\omega\epsilon^* = \omega\epsilon'' + j\omega\epsilon = \omega\epsilon(\tan\delta + j).$$

When the complex resistivity is required in place of the complex conductivity

$$z_v = 1/y_v = r_v - jx_v$$

$$r_v = \frac{g_v}{|y_v|^2} = \frac{g_v}{g_v^2 + b_v^2};$$

$$x_v = \frac{b_v}{|y_v|^2} = \frac{b_v}{g_v^2 + b_v^2} \quad .. \quad .. \quad .. \quad (3)$$

and in the reversed direction

$$g_v = \frac{r_v}{|z_v|^2} = \frac{r_v}{r_v^2 + x_v^2};$$

$$b_v = \frac{x_v}{|z_v|^2} = \frac{x_v}{r_v^2 + x_v^2} \quad .. \quad .. \quad .. \quad (4)$$

$$\tan\delta = \frac{r_v}{x_v} = \frac{g_v}{b_v} \quad .. \quad .. \quad (5)$$

Fig. 2 shows tanδ as a function of the frequency. At frequencies lower than f_1 the main losses are due to ohmic resistance and the curve is of hyperbolic shape. Between f_1 and f_2 the resistance losses drop, but absorption losses rise, resulting in almost constant tanδ.

Beyond f_2 absorption losses rise and tanδ increases with the frequency.

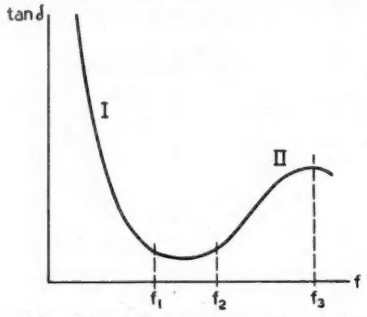


Fig. 2. **Curve showing tanδ as a function of frequency.**

When drying is carried out by means of medium waves the first branch of the curve is applicable, i.e. $f < f_1$.

Figs. 3 and 4 show the dependence of the components of conductivity on the moisture content of birch and pine wood at a temperature of 100° C for a frequency of 250 kc/sec.

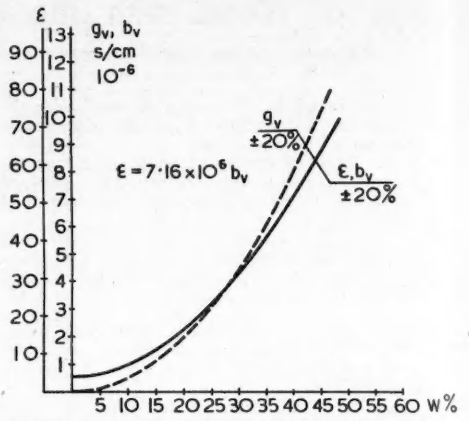


Fig. 3. **Average values ϵ, g_v, and b_v for birch as functions of its moisture content at a temperature ϑ = 100 deg. C and a wave length λ = 1200 m.**

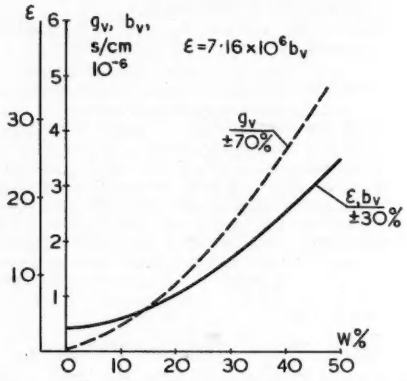


Fig. 4. **Average values ϵ, g_v and b_v for pine as functions of its moisture content at a temperature ϑ = 100 deg. C and a wave length λ = 1200 m.**

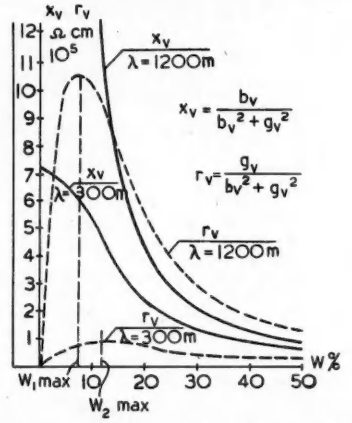


Fig. 5. **Average values of x_v and r_v for pine as functions of its moisture content construed from the curves in Fig. 4.**

The curves are based on average values obtained from large numbers of experiments, the results of which differ from each other by as much as 70 per cent.

For the design of the generator it is necessary to know the specific ohmic and reactive resistances of the load as determined by eqs. (3) and (4). For pine these values have been calculated and are given in Fig. 5. The complex component can easily be compensated by the variometer of the generator load circuit and consequently the ohmic component of r_v mainly determines the design of the generator.

The resistance of a beam can be calculated from the following formulae:—

$$Z = z_v \frac{d}{s}\ ;\ Y = y_v \frac{s}{d} \qquad .. \quad (6)$$

where d is the distance between electrodes and s is the cross-section of wood at right angles to the flow of current.

In calculating the total resistance, the contact resistance between electrodes and wood must not be omitted.

UNEVEN HEATING OF WOOD.

Uneven heating may be due to two main causes:—

(1) The structural properties of wood, mainly a lack of homogeneity.

(2) The varying moisture content of wood.

The second cause is named the thermo-selective effect and is considered in the following.

Fig. 6. **Sketch of series connection of two samples of wood.**

Fig. 7. **Sketch of parallel connection of two samples of wood.**

Let two samples of wood with moisture contents W_1 and W_2 be connected between the electrodes of a generator having a voltage U (Fig. 6). The characteristic constants of the two samples are r_{v1}, r_{v2}, x_{v1} and x_{v2}. With the aid of the variometer and condenser in the load circuit a state of resonance can be maintained, leaving only a resistance load; the output current will then be

$$I = \frac{U_a}{r_{v1}\, d_1 + r_{v2}\, d_2}$$

and the current density

$$\mathfrak{J} = \frac{I}{s} = \frac{U}{r_{v1}\, d_1 + r_{v2}\, d_2}.$$

The power generated in a unit of volume for the first and second sample will be

$$\left.\begin{aligned} p_{v1} &= \mathfrak{J}^2 r_{v1} = \frac{U^2 r_{v1}}{(r_{v1}\, d_1 + r_{v2}\, d_2)^2} \\ p_{v2} &= \mathfrak{J}^2 r_{v2} = \frac{U^2 r_{v2}}{(r_{v1}\, d_1 + r_{v2}\, d_2)^2} \end{aligned}\right\} \quad .. \quad (7)$$

These formulae, in conjunction with the graphs in Fig. 5, lead to the following deductions:—

While the moisture content of both samples is higher than W_{max}, the wood with the lesser moisture content will absorb more heat. On reaching W_{max} this wood will generate less heat, whereas the sample with the higher moisture will generate more until at a certain $W < W_{max}$ equilibrium is obtained.

Higher frequency increases the heating intensity at the expense of r_v. The capacitance of the gap between electrodes and wood can be compensated by the variometer and does not become effective.

When two different samples are connected in parallel, the current density in each is different depending on the air gap (Fig. 7). The current density in the first sample is

$$\mathfrak{J}_1 = \frac{U_a}{\sqrt{(x_{v0}\, d_0 + x_{v1}\, d)^2 + (r_{v1}\, d)^2}}$$

and in the second

$$\mathfrak{J}_2 = \frac{U_a}{\sqrt{(x_{v0}\, d_0 + x_{v2}\, d)^2 + (r_{v2}\, d)^2}}$$

where U_a is the potential between electrodes and $x_v = \dfrac{1}{\omega \epsilon_0}$ is the specific resistance of air.

The power generated in the unit volumes of the two samples is

$$\left.\begin{aligned} p_{v1} &= \mathfrak{J}_1^2 r_{v1} = \frac{U_a^2 r_{v1}}{(x_{v0}\, d_0 + x_{v1}\, d)^2 + (r_{v1}\, d)^2} \\ p_{v2} &= \mathfrak{J}_2^2 r_{v2} = \frac{U_a^2 r_{v2}}{(x_{v0}\, d_0 + x_{v2}\, d)^2 + (r_{v2}\, d)^2} \end{aligned}\right\} \quad (8)$$

When the air gap is reduced the difference between p_{v1} and p_{v2} becomes smaller, and for $d_0 = 0$ eq. (8) becomes:—

$$\left.\begin{aligned} p_{v1} &= \frac{U_a^2 r_{v1}}{d^2 (x_{v1}^2 + r_{v1}^2)} = \frac{U_a^2 g_{v1}}{d^2} \\ p_{v2} &= \frac{U_a^2 r_{v2}}{d^2 (x_{v2}^2 + r_{v2}^2)} = \frac{U_a^2 g_{v2}}{d^2} \end{aligned}\right\} \quad .. \quad (8a)$$

When there is no air gap, the power generated in two samples of wood connected in parallel becomes proportional to their conductivity and, as wood with higher moisture content will possess higher conductivity, its heating will be more intensive.

Although higher frequency contributes to a more equal distribution of heating, it is not desirable in view of the lower efficiency of the generator. The more intensive heating process is detrimental to the mechanical properties of wood, and for large sections of wood a wave band of 300-1,200 m has been found most suitable.

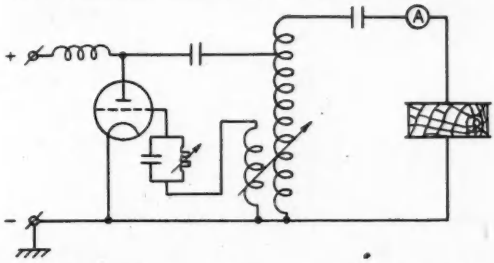


Fig. 8. **Wiring diagram of the generator with a load oscillatory circuit.**

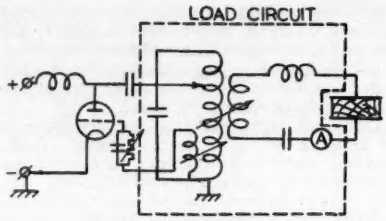


Fig. 9. **Wiring diagram of a generator with an intermediate oscillatory circuit.**

HIGH FREQUENCY GENERATORS FOR THE DRYING OF WOOD.

The principal requirement for a h.f. generator will be its ability to maintain the required temperatures within narrow limits. This is made more difficult by the change in the load characteristics in the course of the process.

Two methods are being used at present:—

(a) generators with load connected in the oscillatory circuit (Fig. 8).

(b) generators with intermediate oscillatory circuit (Fig. 9).

With the first method, the temperature and efficiency are regulated by means of the anode connection and coupling back to the supply. This can only be done when the load is switched off.

With the second method, in addition to the coupling back to the supply, the coupling to the load circuit can also be varied. This method is more elastic in practice, but the set-up is more complicated and includes a large number of condensers.

High frequency generators for the drying of wood are made up of four main components:—(1) a supply cubicle; (2) a rectifier cubicle; (3) generator valves; (4) an oscillatory circuit.

Table I gives characteristic data for two installations of 10 kW and 50 kW output.

TABLE I.

Technical characteristics of installation	Generator 10 kW	Generator 50 kW.
Maximum h.f. output ..	10	50
Rating of anode transformer in kVA	20	84
Number and type of generator valves	4 (GK-3000)	2 (GDO-30)
Frequency kc/sec... ..	330-1000	330-1000
Efficiency (overall) ..	0·46	0·55
Load on generator in m^3 ..	0·4	2·0
Output in m^3/24 hours ..	1·2	4-6
Construction	Panel type	Cubicle type

FRANCE

Test to Destruction of an Electric Motor with Windings of Aluminium Insulated by Electrolytic Oxidation

(From *Revue de l'Aluminium*, Vol. 25, No. 144, May, 1948, p. 175, 1 illustration.)

AN interesting test demonstrating the temperature resistant properties of electrolytically oxidized aluminium wires was carried out recently using a stator winding manufactured by the Société des Moteurs Tollim, of Levallois, near Paris.

The wire used was of 0·8 mm diameter, and had been subjected to an anodic oxidizing treatment followed by immersion in diluted polymerized varnish. The motor was an asynchronous three-phase induction motor with a rating of 1·2 hp at 1,450 rpm, working on 125 volts.

Certain precautions were taken in making the stator sections to avoid scaling or expansion effects. The stator sections were machine-wound at a slow speed on rectangular forms with rounded corners. Furthermore, care was taken to avoid crossing two wires entering and leaving a coil.

The stator was provided with a rotor in short-circuit, with a cage which was a 95·5 per cent aluminium casting, and two ordinary covers. The whole motor, in fact, was made of aluminium, with the exception of the shaft, ball bearings, and magnetic stampings.

The tests were carried out in three different stages: (1) a normal running test, followed by (2) an overheating test, and (3) an overload test at a high voltage until the motor was put out of use. The first test showed that the motor was functioning satisfactorily; in the second test the motor reached abnormally high temperatures before the experiment was discontinued. After stripping down, it was found that the insulators, consisting of oiled cloth and impregnated cardboard, had begun to change their colour, but the oxidized wire windings of the stator did not appear to have been affected.

These first results were considered sufficiently encouraging to continue the test up to the destruction of the windings, which was to be achieved by raising the supply voltage. Although the normal rated voltage was 127 volts, the motor was then fed with 220 volts, and as a result the current increased from the normal 2·5 amperes at the beginning of this test to 20 amperes, or 8 times its normal value.

After running for five minutes, volumes of smoke began to come out of the motor, and the smoke rapidly became so dense that it was impossible to remain in the test room. At the end of ten minutes flames appeared; in a short while they developed to such an extent that, after about 20 seconds' running under these conditions, the test was voluntarily ended. After cooling down, the motor was stripped for examination. It was certain that the fire was due to the insulators which were ignitable at the high temperatures attained. The oxidized wire windings were blackened by the smoke and the carbonization of the varnish used for impregnation; but the first inspection seemed to indicate that they were not destroyed. In fact, after checking the insulation of each phase, the motor was run again and its functioning was absolutely satisfactory.

This test again confirms the fact that aluminium conductors insulated by anodic oxidation make it possible to construct machines capable of withstanding considerably higher temperatures, either under temporary and abnormal conditions, or under special conditions required for normal operation. Recent progress in the development of insulating materials with high temperature-resistant properties will no doubt enable the electrical industry to take advantage of these possibilities in the near future.

DESIGN AND WORKSHOP

A Digest of articles of interest to Works Managers, Planning and Production Engineers, Superintendents, Jig and Tool Designers, Inspectors, Time Study Engineers, and all those concerned with the practical aspects of Engineering.

Hot-Pressing and Hot-Forging Light Alloys

By M. CHARTRON.

(From *Revue de l'Aluminium, France*, Vol. 25, Nos. 141 and 143, pp. 37-43, and 113-122, 27 illustrations.)

(*Continued from September issue*).

IV. RULES FOR THE PLASTIC DEFORMATION OF METALS

A. Tresca's Laws

IF blocks of the same material and of equal weight are compressed on a press of a certain capacity until, due to cooling or work-hardening, further compression is impossible, the following laws apply:—

(1) The final depth of the deformed piece is dependent neither on the initial height of the billet nor on its shape and cross-sectional area.
(2) The required final pressure per unit of surface area of the deformed piece depends on the ratio of the final diameter to the final depth.
(3) If billets of different shapes are pressed until they become disks of equal final diameter, the required final pressure varies inversely with their final heights.
(4) If different billets are pressed until they become disks of equal final depth, the final pressure varies in linear relation to the final diameter obtained.

Experience shows that the pressure is, generally, a linear function of the ratio of the final diameter to the final depth of the deformed piece (2nd law) and its value can be found from:—

$$p = k + m\,D/h = k\,(1 + m\,D/k\,h) = k\,(1 + a\,D/h)\,(1),$$

where D = final diameter, h = final depth of disk, m and k = constants, and $a = m/k$. Tests on semi-hard steel, brass and duralumin showed that the pressure is constant for the same material and equal ratios D/h, and varies only with the working temperature. Tests with varying degrees of polish and lubrication of anvil and tup surfaces proved that the constant k depends only on the type of metal and its temperature, whilst the constant a depends on the surface polish and lubrication of the anvil and tup surfaces. Fig. 2 gives results of tests on duralumin billets and makes it possible

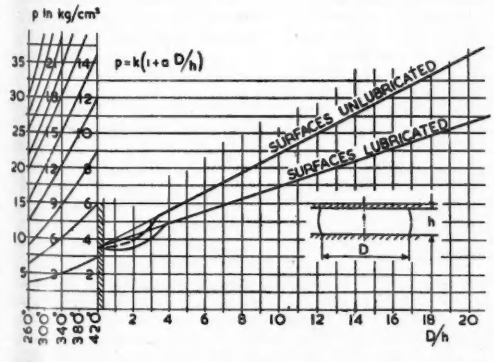


Fig. 2. **Pressure required for the compression of duralumin disks as a function of the ratio D/h, forging temperature, and the structure of tup and anvil surfaces.**

to determine the final pressure for different ratios D/h, different working temperatures, and different surface structures of the anvil and tup surfaces. Only two curves are shown, but by varying the lubrication and surface finish, more curves could be obtained. Tests also verified Tresca's third and fourth law. Siebel* gives a very similar formula from theoretical considerations:—

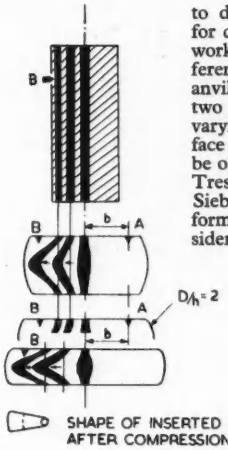


Fig. 3. **Flow of metal during compression of a disk.**

$$p = k_f \left(1 + \frac{2}{3}\,\mu\,\frac{r}{h}\right) \quad \text{where}$$

k_f = resistance to plastic deformation of a unit cube of metal, μ = coefficient of friction between the work piece and tup and anvil surfaces, r = radius, and h = depth of disk.

B. Range of Application of Tresca's Laws

Tresca's laws, eq. (1), are not valid for small values of D/h (see Fig. 2), and to clarify conditions for these, the following tests were made. Axial holes were drilled in drawn duralumin bars, and then filled with drawn rods of the same material (Fig. 3). The bars were then pressed to different heights and the rods carefully removed by cutting. Results showed that

(1) only the central rod keeps a straight axis but its diameter is enlarged near the centre, and more so as the total deformation of the bar increases.
(2) all other rods buckle near the centre ;
(3) during the first part of the compression no change either in rod diameter or in rod centre distance is noticeable from the outside ;
(4) as the compression increases, a point B marked on the surface of the original cylindrical bar approaches the flat cylinder base, but when it reaches the base its distance from the disk centre remains constant ;
(5) When $D/h = 2$, point B begins to move away from the disk centre, the required pressure begins to increase more rapidly and the perimeter of an axial cross-section through the disk (as shown in Fig. 3) receives its minimum length.

Three ranges of compression can thus be distinguished :—

*Siebel: *Plastic deformation of metals*, Ch. Beranger, 1936.

(a) Start of compression. The cylinder surface is rolled into the base plane without friction. The required pressure is independent of the surface of the anvil.
(b) Advanced compression. The surfaces of tup and anvil and the ratio D/h of the compressed disk strongly influence the required pressure. Tresca's law, eq. (1), is obeyed in this range.
(c) Intermediate range, when D/h is approximately 2. (See Fig. 4).

These characteristic ranges are also distinguishable on the graph represented by Fig. 2. Below $D/h = \frac{1}{3}$, no compression is possible without buckling of the bar. A range of relatively constant pressure p then follows, with a gradual transition at about $D/h = 2$ to the final range of Tresca's laws. The transition range is not sharply defined.

C. Importance of Tresca's Laws

To determine the necessary pressure p for a given material under certain conditions, a given press must deform two test billets of different weight and when they cannot be deformed further by the press, D/h should not be less than 2. These two tests determine the coefficients k and a and make it possible to calculate the necessary pressure, under any given conditions, for the tested material. When this pressure is known, the energy required for the forging of a given piece, with the anvil width required in a press of a certain capacity, and a maximum safe value of D/h to obviate danger of breakage, can be found. The overheads chargeable to forge presses of different capacity vary appreciably, and the price of forgings can be reduced by using a smaller yet sufficiently powerful press. Customers can be advised on the choice of mechanically similar materials which require different forge pressures p, and on the best shape for their forgings.

D. Hot Die Pressing

Three compression ranges can also be distinguished for die pressings (Fig. 5) ; firstly a range AB during

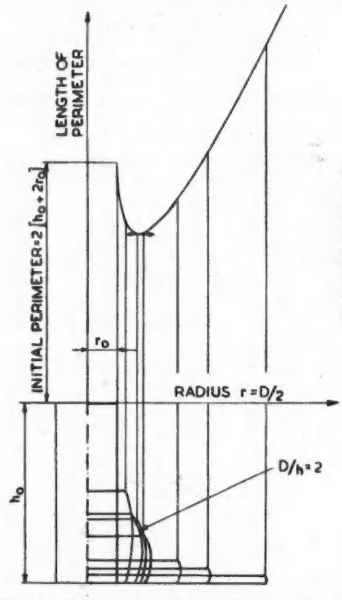


Fig. 4. **Change of perimeter length of an axial cross-section through the disk as a function of the disk radius at various stages of compression.**

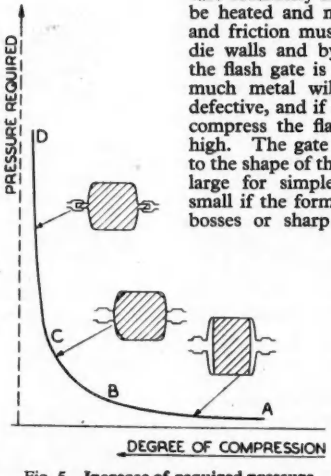


Fig. 5. **Increase of required pressure during various stages of hot die-pressing (drop forging).**

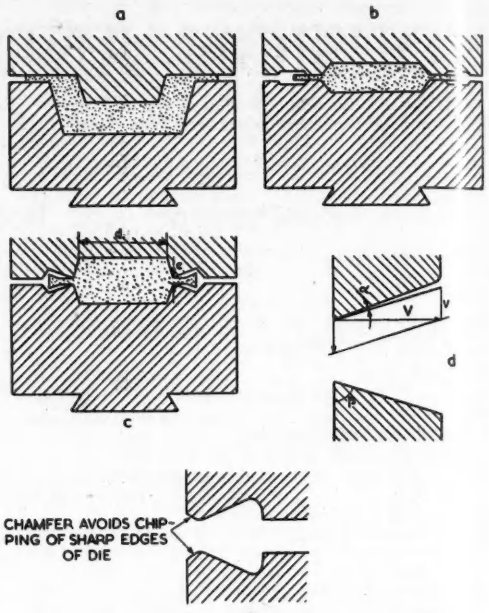


Fig. 6. **Different shapes of flash gates.**
v = speed of die movement during compression, V = speed of metal flow. $\tan \alpha > v/V$.
For cylindrical disks, $\frac{\pi d^2}{4}\, v = \pi de\, V$; therefore, $\tan \alpha > \frac{4e}{d}$.

which the billet can still be expanded outwardly in the die. The conditions are the same as for compressing between plates, and as soon as the billet touches the vertical die walls, the pressure required increases rapidly (BC). Finally, the pressure builds up as soon as flash is formed, and a very small additional compression requires an extremely high pressure. To limit this pressure, the billet must be introduced into the die at the maximum permissible forging temperature accurately measured by a pyrometer, the dies must be heated and maintained at a sufficient temperature, and friction must be reduced by fine surface finish of die walls and by adequate lubrication. The form of the flash gate is very important. If it is too large, too much metal will run out and the pressing will be defective, and if it is too small, the pressure required to compress the flash material will be disproportionately high. The gate opening should, therefore, be adapted to the shape of the work-piece and make it comparatively large for simple pressings of normal thickness, but small if the form is complicated and has thin passages, bosses or sharp angles. The exact form should be determined by tests. Fig. 6 shows a simple flash-type die as is often used when the form is entirely worked out of the lower die and is relatively simple. Fig. 6b is an improved form which limits the flash area under pressure and, therefore, the maximum pressure required at the end of the forging process. The gate Fig. 6c to e is more expensive, but reduces as much as possible the maximum pressure required for die-pressing a given piece.

(To be continued.)

New Principle Employed in Gear Checking Machine

(From *Machine Design, U.S.A.*, Vol. 20, No. 6, June, 1948, pp. 128-130, 5 illustrations).

EMPLOYING a new type gauging element known as a master worm section, the machine shown in Fig. 2 checks all types of spur and helical gears. The master worm section is based on the same principles as a hob, Fig. 1, and is ground to extremely close tolerances.

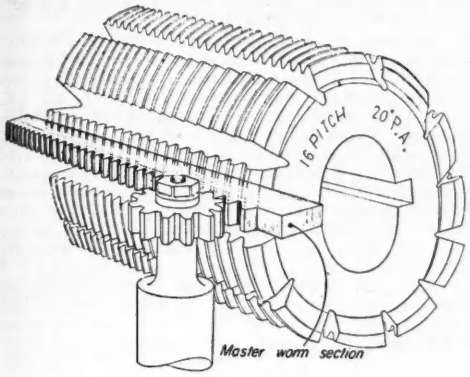


Fig. 1. **Diagram demonstrating that master worm section is based on same principles as a hob.**

The machine, a development of the Eastman Kodak Company, is of the variable-centre-distance type in which the master worm section is traversed longitudinally in engagement with the gear being tested, Fig. 2. Errors in the test gear such as departure from theoretical size, profile error, tooth-thickness error, tooth-spacing error, and eccentricity or run-out are manifested as variations in the centre distance. An electronic pick-up records these variations on a moving chart providing a record of the composite error on the gear. From this chart individual errors may be evaluated.

In designing machines to make use of the concept of the worm section, two major problems were encountered. First, it was necessary to ensure that the worm section travel in a straight line. Second, the inertia of the work table had to be such that its sensitivity approximated that of the electronic pick-up.

First of these problems was met by mounting two hard-faced vertical rails to form a V-way in which the carriage holding the master worm section rides against recirculating ball bearings. One of these rails is indicated in Fig. 3. These rails are of laminated construction with a front surface of high-carbon steel hardened to Rockwell C 67. A massive, box-type casting supports them, ensuring rigidity.

Each rail defines a plane and the inter-section of two planes is, by definition, a straight line. The worm section is traversed in a line parallel to the line formed by these planes.

A spring-loaded, recirculating ball-bearing bushing, Fig. 3, moving up and down a vertical rod between the rails, acts as a retainer to hold the carriage firmly against the V-way. The floating action of this bushing eliminates any tendency to impart direction to the travel of the carriage. The position of this bushing, and the position at which the driving chains are attached to the carriage, have been selected so that the load on the bearing surfaces is equalized.

Two chains, passing over a sprocket at the top of the instrument, connect this carriage to a counterweight, Fig. 2. A 1/20-hp synchronous gear-motor, driving through an overload-release clutch, is used to drive the master worm section at a speed of 2 ips. This enables

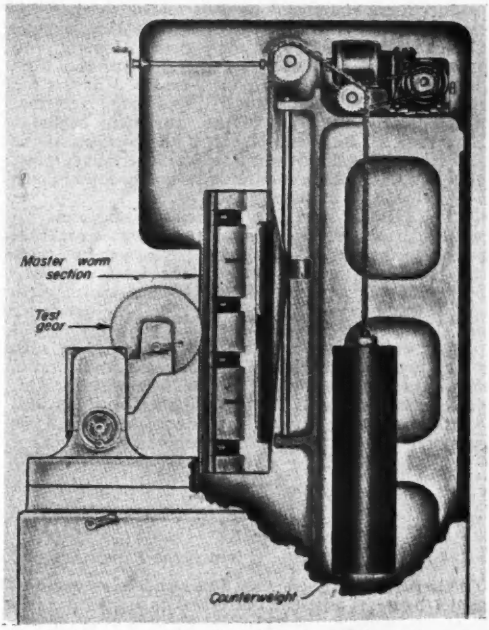


Fig. 2. **Cut-away section of machine showing how the master worm section is traversed longitudinally in engagement with the gear being tested.**

the operator to check a gear of 8¼-in pitch diameter—the maximum capacity of this machine—in 13 seconds. A cycle switch automatically stops the worm section when it has travelled the distance necessary to effect a complete revolution of the gear under test.

In designing the work table, maximum sensitivity was assured by mounting the carriage on 64 selected balls. Half of these move in a double vee on one side of the carriage while those on the opposite side are mounted on a vee and a flat. This eliminates the need of parallel vees at each side of the table.

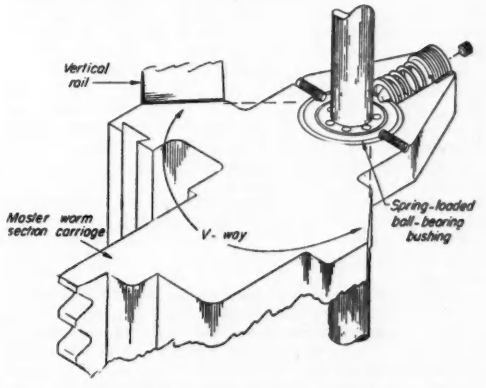


Fig. 3. **Master worm section carriage rides on vertical V-ways and spring-loaded ball-bearing bushings.**

All raceways are of similar construction to that of the vertical rails. Inertia is such that the carriage responds to 40 cps for 0·001-in of travel. This results in the exploration of numerous points on all tooth profiles.

Electronic equipment used to measure centre distance variation is made by the Brush Development Company and consists of three principal parts: An induction pick-up, located at the front of the work table; a displacement amplifier, mounted in the top right of the cabinet; and a direct-inking recorder, located at the top front of the instrument behind a glass pane. Lateral movement of the carriage as the gear is rotated against the master worm section is transmitted to the feeler arm of the pick-up head, producing a signal which is amplified and recorded.

The pick-up head is a differential transformer in which the centre coil moves between two fixed coils wound so that their fields oppose. When the movable coil is exactly centred between the fixed coils, the resultant e.m.f. is zero. However, when the movable coil is nearer to one end than the other, the unbalance produces a voltage.

Output from the movable coil is rectified in such a way that the direction of displacement from centre, as well as its magnitude, can be determined. This demodulated signal is further amplified and recorded on the paper tape. Tapes can be made of the same gear and any one exactly superimposed over another, since both the recorder and the master worm section are driven synchronously.

Sensitivity of the recorder is such that mechanical magnifications up to 800X can be used. This corresponds to approximately 0·00005-in per chart division. A knob on the front of the instrument permits the use of selective magnifications of 100X, 200X, 400X, or 800X. In addition, three tape speeds are available. When inspecting between overall tolerance limits a low tape speed is desirable so that run-out is clearly separated from tooth to tooth flicker.

Another feature of the work carriage is a built-in cross-traverse slide which permits a gear to be traversed at right angles to the master worm section. Any taper which exists in the gear is recorded when the gear face is explored in this way. Amount of cross-travel is measured by a scale and handwheel with a dial graduated in thousandths of an inch. This feature also allows the crowning of a gear to be checked in the same manner as taper after the face of the gear has first been explored to find the high point of the crown. Moving a gear a known distance and noting the new indicator reading provides data for simple calculation to determine amount of crown.

Also built into the work carriage is an accurate sector, on which the work centres are mounted. This permits rapid positioning of the work gear axis to any helix angle up to 45 degrees either hand.

Shop-Run Tolerances

By L. M. NIELSEN. (From *Product Engineering, U.S.A.*, Vol. 19, No. 6, June, 1948, pp. 141-145, 7 tables.)

THE variation in tolerances, finish and other requirements between different operations make it advisable to set up shop-run tolerances for screw machine operations. Usually engineers and inspectors follow a particular line of work and explain the functional requirements or finish. Shop-run tolerances indicate specifications that avoid much confusion.

The eccentricity tolerances within which various types of parts can be maintained economically on the usual types of screw machines are shown in Table I. The angularities between thread and body, or thread and thread are shown in Table II. These limits apply on work up to 10-in long, which usually covers a majority of the parts made on screw machines.

Manufacturers strive to produce parts as close to zero variation as possible, but wear and tear on machines, tools, and fixtures as well as the human element makes it necessary to plan for some variations. When inspectors find a tendency for the work to approach shop-run tolerance limits, the maintenance man or the tool setter can be called in to make the necessary corrections. Tables III and IV are the variations to which the preceding comments will apply. The squareness of faces under bolt heads, screws, and nuts, shown in Table IV, is important mainly where high stress is encountered, but it is advantageous to specify what may be expected on commercial work so that a surface requiring closer attention may be so indicated on the drawing. The tolerances given in Table III for turning, boring, and, facing operations can be used on hand screw machines, chucking machines, and second operation parts. These tolerances are given for dimensional variations where much misunderstanding can occur.

The reliefs used where threads are close to a shoulder or run into a blind hole that cannot be drilled deep enough to provide clearance for chips are shown in Table V. These also apply where a bolt or screw extends deeper than perfect threads would permit. Tapping such holes, even with a bottoming tap, is difficult without chip clearance. This practice is not used on holes less than $\frac{1}{4}$-in diameter. The drilled hole should be deep enough to allow for imperfect threads and provide space for chips.

Certain other standard practices should be defined so that they may be mutually understood. The following are typical examples:

Sharp Corners. Unless otherwise stated on the drawing, these should be "burred." Practical limits for this are: Up to 1-in dia., 0·003-in to $\frac{1}{64}$-in; to 3-in dia., 0·003-in to $\frac{1}{32}$-in, and over 3-in dia., 0·003-in to $\frac{1}{16}$-in, all on a 45-deg angle.

Chamfers, Internal and External. Chamfers require an additional operation and should not be confused with "break corners" or "burr." When chamfers are required they should be specified on drawings by sizes as: $\frac{1}{16}$-in chamfer. The angle of chamfer can be standardized, such as 45 degrees unless otherwise specified. Exceptions can be made so that existing tools or equipment will not be affected. Bolt heads and nuts, for example, can be made to a different standard.

Chamfers on Threaded Ends. Chamfers, unless otherwise specified, can be made to extend below root diameter of thread, and the depth below the root diameter specified shall not exceed one-half the depth of thread. Squared ends with chamfered edges are preferable to rounded ends, since the rounded ends require form tools and radius gauges.

Corner Fillets. When a turned diameter runs into a larger diameter a square corner is hard to maintain and presents a definite stress raiser, or a weak point. It should not be used unless the design requires it. A radius is always preferred. Unless otherwise specified, corner radii can be made to a radius of $\frac{1}{64}$-in for diameters up to perhaps $\frac{3}{4}$-in and $\frac{1}{32}$-in radius for larger diameters. Since fillet dimensions may be critical, they are usually specified on the drawing.

Flatness of Ends. Cut-off ends can be expected to vary as either concave or convex to approximately 0·003-in max., per inch of diameter; and faced ends to 0·001-in max. per inch of diameter.

Class Thread. All threads, either external or internal

TABLE I. ECCENTRICITY TOLERANCES FOR SCREW MACHINE WORK

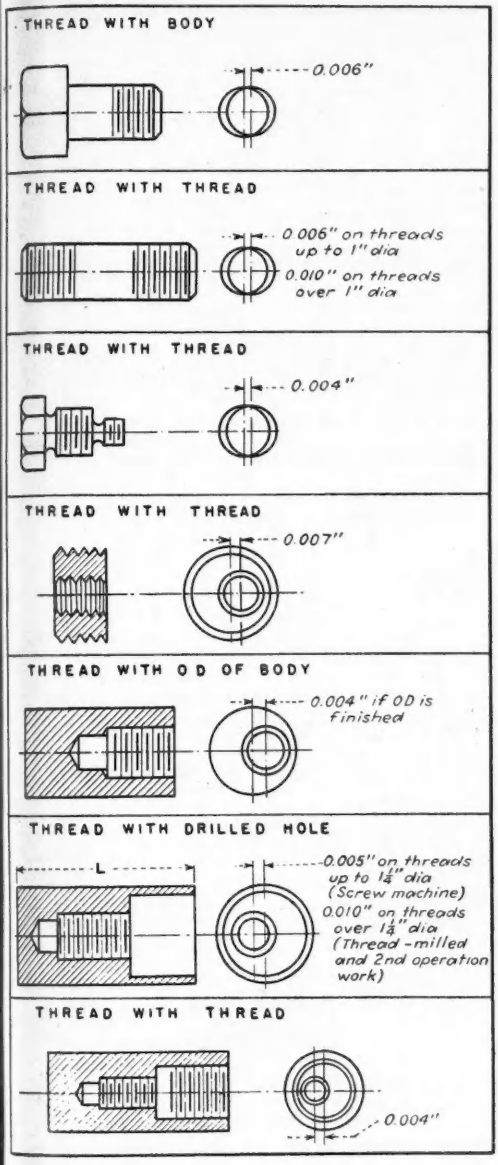
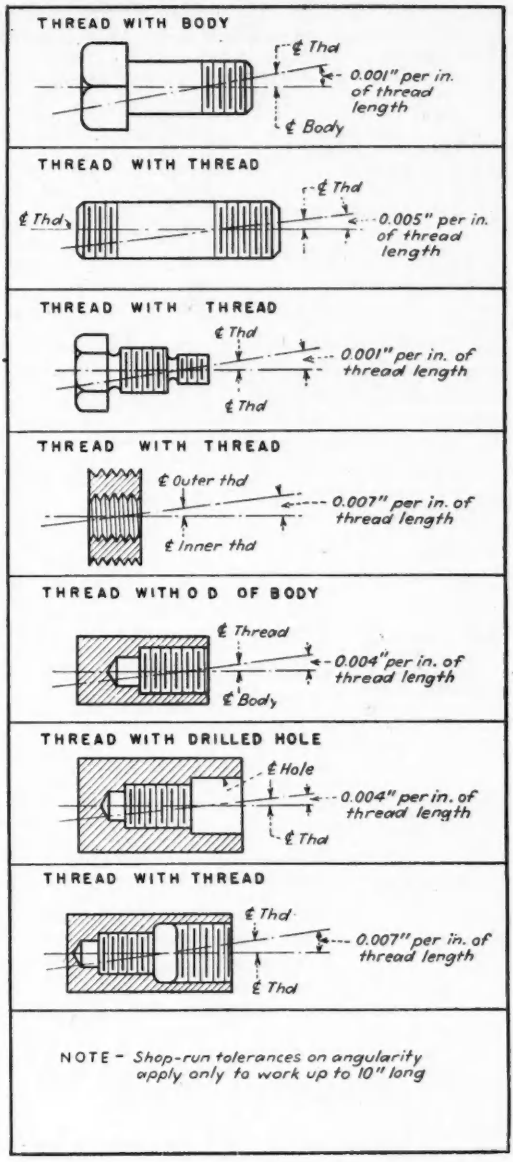

THREAD WITH BODY
0.006"
THREAD WITH THREAD
0.006" on threads up to 1" dia
0.010" on threads over 1" dia
THREAD WITH THREAD
0.004"
THREAD WITH THREAD
0.007"
THREAD WITH O D OF BODY
0.004" if OD is finished
THREAD WITH DRILLED HOLE
L
0.005" on threads up to 1¼" dia (Screw machine)
0.010" on threads over 1¼" dia (Thread-milled and 2nd operation work)
THREAD WITH THREAD
0.004"

TABLE II. ANGULARITY TOLERANCES FOR SCREW MACHINE WORK



can be made to conform to a specified class unless otherwise indicated on the drawing.

Grinding Relief on Shafts. When a ground diameter runs into a larger diameter, a relief groove near the face of the larger diameter should be provided. Unless an undercut is shown on the drawing, these are practical dimensions: Width, 0·030-in max.; depth, 0·003-in to 0·005-in or 0·006-in to 0·010-in diameter below the finished diameter called for on the drawing. The radius in such cases can be 0·004-in to 0·007-in near the face from the larger diameter, and leave at an angle of 8 deg to 12 deg.

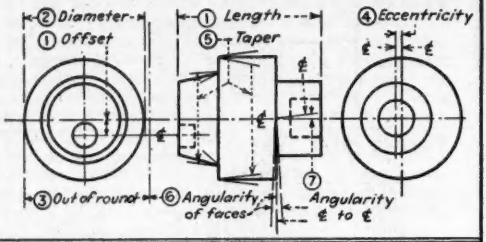
Tolerance for Holes. Tolerances for holes drilled as part of screw machine operations are shown in Table VI.

GENERAL MACHINE SHOP

Some shops building aviation and marine equipment require close-tolerance production, while others such as builders of industrial control equipment use a large number of moulded and punched parts on which wider tolerances are acceptable. Also, there are many shops that have a mixture of requirements for both close and wide tolerance work.

Holes that can be drilled, punched, swaged, etc.,

TABLE III. GENERAL TOLERANCES FOR SCREW MACHINE WORK



①	Length	0" to 1/4"	Over 1/4" to 24"	Over 24" to 72"	Over 72"	
	Tolerance	±0.008"	±1/64"	±1/32"	±1/16"	
②	Diameter	0" to 1/4"	Over 1/4" to 24"	Over 24" to 72"	Over 72"	
	Tolerance	±0.008	±1/64"	±1/32"	±1/16"	
③	Out of Round	For all diameters, these factors are to be within the tolerance on diameter, whether the tolerance is on the detail drawing, or the shop-run tolerance on diameter applies				
	Tolerance					
④	Eccentricity					
	Tolerance					
⑤	Taper					
	Tolerance					
⑥	Angularity	Faces and shoulders				
	Tolerance	Max 0.005" per ft of surface width measured radially				
⑦	Angularity	Centre line to centre line				
	Tolerance	Max 0.010" per ft				

The dimensional tolerance may not be exceeded by any combination of other tolerances affecting the same surface or surfaces

TABLE IV. BEARING FACE ANGULARITY TOLERANCES FOR SCREW MACHINE WORK

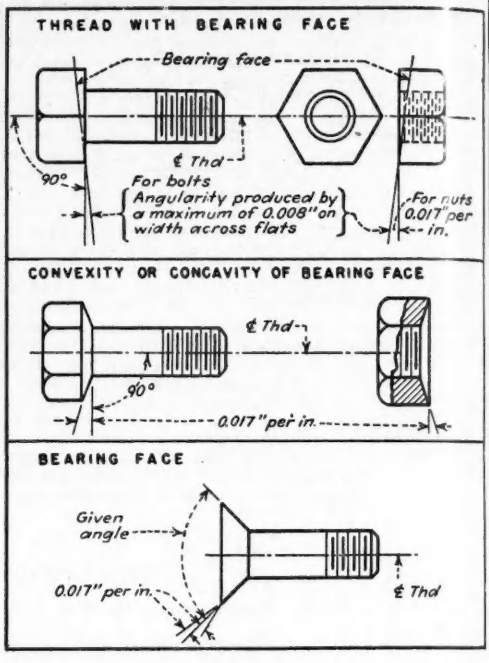
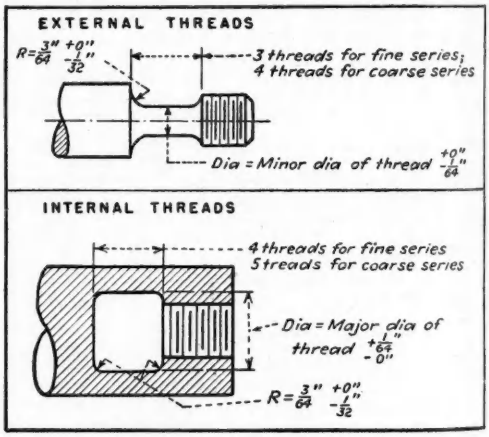


TABLE V. THREAD RELIEFS FOR SCREW MACHINE WORK


EXTERNAL THREADS
R=3/64" +0" -1/32"
3 threads for fine series; 4 threads for coarse series
Dia = Minor dia of thread +0" -1/64"
INTERNAL THREADS
4 threads for fine series
5 treads for coarse series
Dia = Major dia of thread +1/64" -0"
R=3/64" +0" -1/32"

can have the word "hole" follow the dimension on the drawing, permitting the method of machining to be determined by the factory. This same principle can be observed wherever alternate methods of production may be used. The factory can thus select the tools and methods, best adapted to both its available facilities and past experience, and produce a satisfactory part at the lowest possible manufacturing cost.

Typical shop-run tolerances on circular bodies indicating allowable variations for: Out-of-round, eccentricity, taper, angularity of faces and shoulders to the centre line and the angularity of diameters with the centreline are similar to those given in Table III. Only the values for angularity tolerances are different from those shown in Table III. For general machine shop operations the maximum angularity tolerance for faces and shoulders is 0·003-in per ft of surface width measured radially. The maximum centre line to centre line angularity tolerance is 0·003-in per foot. Angularity of diameters should not be confused with taper, as the diameter may measure the same on both ends, but may run true with the centre line at only one point along the axis. Also, as pointed out in Table III, the combined variation in eccentricity, out-of-round, and taper may not exceed the tolerance on the diameter.

The following values for run-out can be used for holes in the centre of parts (run-out being twice the eccentricity) produced on different types of machine tools:

TABLE VI. TOLERANCES FOR HOLES DRILLED ON SCREW MACHINES

Basic Hole Diameter	Variation from Basic Size in inches			
	Metallic*		Non-Metallic**	
	Plus	Minus	Plus	Minus
Up to 0·040-in	0·002	0·001	0·001	0·002
0·040-in to 0·070-in ..	0·003	0·001	0·001	0·003
0·070-in to 0·128-in ..	0·004	0·001	0·001	0·004
·128-in to 0·228-in ..	0·005	0·002	0·002	0·005
1/4-in to 1/2-in ..	0·006	0·002	0·003	0·006
1/2-in to 3/4-in ..	0·008	0·002	0·003	0·008
3/4-in to 1-in	0·010	0·003	0·005	0·010
1-in to 3-in	0·015	0·003	0·005	0·015
3-in and over	0·020	0·005	0·007	0·020

*Metals, Mycalex or similar non-shrinkable materials.
**Non-metallic materials having shrinkable characteristics.

Drills—not to exceed 0·010-in between outside surfaces and centre.

Turning and Boring Machines—not to exceed 0·007-in between outside surfaces and centre.

Lathes—not to exceed 0·005-in between outside surfaces and centre.

Grinders—not to exceed 0·001-in between outside surfaces and centre.

Facing should not be convex or concave more than 0·001-in per inch of diameter.

Cylindrical parts with thin walls, thin walled rings, half-rings or shells which, unless measured in the fixture in which they are machined, will go out of shape until assembled with the mating parts frequently present problems. To study such situations, the following methods can be used to arrive at a satisfactory solution.

Where the ratio of inside diameter to the wall thickness dimension is 10 or greater, special consideration must be given to distortion of diameters in making the set-up for machining and clamping, and later measuring. Average diameters may be used to measure the diameters of such parts after all clamping has been released. Six or more measurements can be taken and averaged to obtain average diameter. Outside diameters of such parts can also be measured by flexible rules around the body, dividing the measurement by π.

Since the shop-run tolerances for individual operations are constantly being decreased by improved manufacturing machinery and methods, the standards have to be revised to suit these varying conditions.

Shop-run tolerances for other specialized manufacturing operations such as forging, broaching, welding, etc., can be developed to suit the requirements of the particular organization. For example, studies made on flame-cut and welded parts can be the basis for shop-run tolerances on dimensions, angles, etc., as well as on the surface finish of flame-cut and welded seams. Allowable variations for this type of work depend upon the size of the pieces used. Often it is advantageous to take photographs of welded or flame-cut surfaces to indicate the general appearance of acceptable work.

Carbide Drills

By F. W. LUCHT. (From *The Iron Age, U.S.A.*, Vol. 161, No. 25, June 17, 1948, pp. 88-90, 5 illustrations.)

HIGH production rates in drilling are possible with carbide drills because they permit the use of heavier feeds ; give long drill life between grinds ; perform well on holes where a chip problem may develop if a twist drill is used ; maintain accuracy ; produce smooth holes with a minimum of run-out ; permit drilling larger holes from the solid instead of removing material by successive drilling operations ; and will drill holes much closer to desired size.

Carbide-tipped all-depth drills may be divided into two types : (1) the solid carbide type, shown in Fig. 1, and (2) the wear strip types shown in Figs. 2 and 3. Drills from $\frac{1}{4}$ to $\frac{5}{8}$-in dia. are ordinarily made to the solid carbide design. In this type, the entire drill end is carbide, copper brazed to a steel shank or body.

The wear-strip type of drill shown in Figs. 2 and 3 is made with one carbide cutting tip and two carbide wear strips.

The brazed-type wear strip drills shown in Fig. 2 can be made from 0·290 to 1-in diameter and the same size drill tubes can be used as are used for solid carbide type drills. Tips are attached with the same brazing method. This drill is easier to fabricate than the solid carbide drill since carbide blanks can be purchased and the complete drill end can be made in the tool room.

The wear-strip type drill shown in Fig. 3 has the advantage of easier handling, particularly during sharpening, since the drill end is detachable. Drills of this type are made in diameters ranging from 1 to 3 in. In larger sizes, the solidly attached end type tend to become heavy and more cumbersome to handle. The shank-end of this drill, known as the wear-strip-detachable type, is an adaptation of Pratt & Whitney drill bars.

The outside diameters of all carbide tipped all-depth drills are ground cylindrical with 0·0006 inch

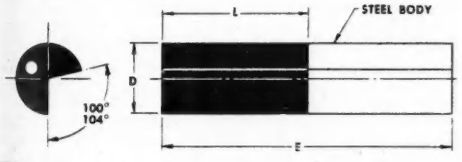


Fig. 1. **Solid carbide type of all-depth drill.**

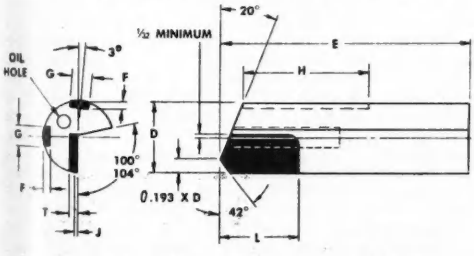


Fig. 2. **Brazed type wear strip all-depth drill.**

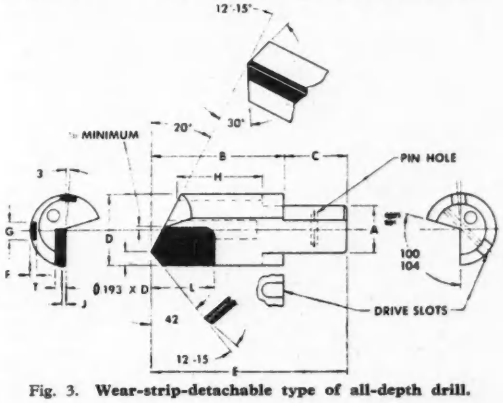


Fig. 3. **Wear-strip-detachable type of all-depth drill.**

per inch included back taper. After the o.d. has been ground, a longitudinal clearance should be ground along the cutting tip, leaving a circular land below the cutting edge. This land, *f* in Figs. 2 and 3, should vary in width from 0·030-in for a 0·290-in dia. drill to 0·050-in for a 3-in drill. The face of the cup-type grinding wheel should just clear the periphery of the steel drill body in grinding this clearance. This land should never be touched when sharpening.

The o.d. of the steel bodies for all wear strip drills should be about 2 per cent smaller than the drill size. The oil hole in the shank end of the wear-strip-detachable type drills is counterbored to a sufficient diameter and depth to receive the copper oil tube. The drill bar should never be longer than is necessary.

The cutting edge of carbide drills should be from 0·003-in below centre to on centre—never above centre—because this produces a freer cutting action. The cutting face side of the drill flute should be located in the drill end about $\frac{1}{64}$-in ahead of the corresponding side of the flute in the drill bar. This prevents the formation of a step to which chips might cling.

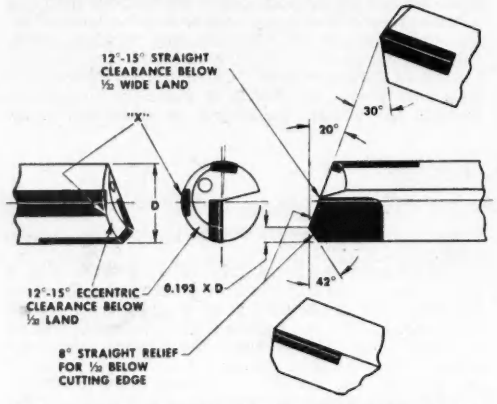


Fig. 4. **Cutting and relief angles for carbide all-depth drills.**

The successful operation of any all-depth carbide drill depends largely on the use and maintenance of correct cutting and relief angles. A drill ground to the specification shown in Fig. 4 will be satisfactory for most drilling operations. All grinding should be done free-hand, and the same type of grinder may be used as is used for carbide tipped single-point tools. When rough grinding the 42° outer cutting angle, the drill should be held firmly by hand in a V-block, which is swung at a 42° angle, and the grinder table tilted at a 12° to 15° angle. During grinding, the drill should be rotated about its own axis, as when grinding any drill.

When rough grinding the 20° inner cutting angle, the drill should be held firmly by hand in a V-block which is swung at a 20° angle and the grinder table tilted at the 12° to 15° clearance angle. A 60 grit silicon carbide cup-type wheel should be used.

Finish grinding both the 42° and the 20° angles at an 8° relief angle can be done in the same fixtures, but the table tilt is only 8°. The land below the cutting edge should be about $\frac{1}{32}$-in wide. To obtain a keen cutting edge, 100 to 150-grit diamond cup-type wheels are used for finish grinding. A light hand feed of about 6 ipm should be used for the final pass along the grinding wheel. Lapping on the cutting edge with a 220 or 240 grit diamond wheel improves operating conditions on some jobs.

Sharpening should be done only on the end of the drill. The face of the cutting tip should never be touched, since this face is located on centre to 0·003-in below centre when the drill is made and should not be altered.

The all-depth drill, because of its design, is self-grinding in the work. Its tendency, therefore, is to follow the direction in which it has been started. Drill guide bushings in the combination drill-guide-and-work-support are important in drill set-up. For best results, the drill should be guided through the bushings and fed into the work while the work is rotating at the correct cutting speed.

Chip removal is the most important function served by the cutting fluid. The fluid is forced under pressure through the entire length of the drill into the annular-shaped groove produced by the drill and out through the V-shaped flutes. This washes the chips through the flute to the open end of the hole and into the chip duct. Oil pressures for various carbide drilling jobs vary between 300 and 800 psi, depending upon the drill size, depth of drilled hole and feed rate. Cutting oils seem to develop a smooth, springy type chip which has a minimum tendency to pack in the drill flute. They also leave a keen cutting edge.

In drilling steel, work should be rotated at a speed that will give a cutting speed at the outside diameter of the drill ranging from 130 to 350 sfpm. A new job should be started at from 200 to 250 sfpm and the speed varied in either direction until a satisfactory balance between drill life and the feed rate is obtained.

Gun drill work to date has been done mainly with feeds within the range of 0·006 and 0·0036 inch per revolution, the lighter feeds for smaller drills and the heavier feeds for larger sizes. Feed rates depend upon oil pressure used and the size of oil holes through the drill. Oil flow must be sufficient so that there is no tendency for chips to pack in the drill flute. Wear strips last longer if small oil grooves are milled ahead of each wear strip, the rear end of each groove being connected to the main oil hole. It has also been found that, as feed rate is increased there is an increased tendency for the drill to deviate from its true course.

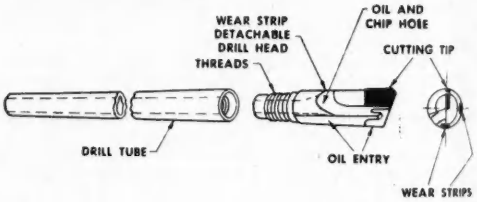


Fig. 5. **This special design of a short drill can be rotated instead of rotating the work. Cutting angles are comparable to those used in stationary drills, but chips are carried out through the centre of the drill instead of through an external axis.**

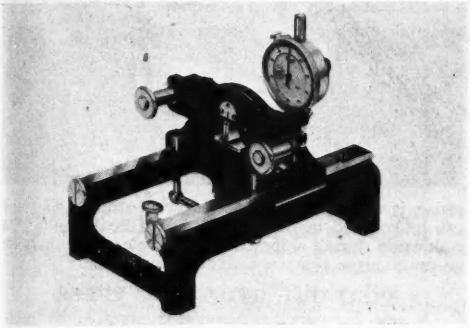
Long drills, generally, should not be rotated. However, if drills such as shown in Fig. 5 are used, it may be feasible to rotate the drill. Cutting angles on this type are comparable to those used in stationary drills, but the chips are carried out through the centre of the drill rather than through an external flute.

A steady rest should be placed about midway between the drill feed slide and the combination drill guide and work support carriage. To eliminate "singing" in a drill, the result of high frequency vibration or chatter, triangular-shaped wooden wedges can be driven into the guide pushings. Current indications are that when all-depth drills approach a diameter of 2·5-in, it is necessary to break the broad chip which comes from both the 20° and 42° cutting angles. This can be done by grinding steps in the cutting edges. A slight change in the relationship between the 42° outer cutting angle and the 20° inner cutting angle may also aid in breaking up the chips flowing from the drill.

New Materials, Processes and Equipment

MICROMETER-CONTROLLED TEAR CUTTER

The instrument is produced by Avimo Ltd., of Taunton, Somerset. It is designed with particular regard to the difficulties experienced in testing sheeting of soft, unloaded rubber compound, and gum stocks. A heavy base is cast in one piece with the machined guide rails along which the sample carriage travels. The rubber sample is clamped in the carriage by means of two spring-loaded disks. A razor blade is held firmly in a clamp, and when the sample carriage is moved along the rails the edge of the blade enters a narrow slot in the carriage and cuts the rubber. The carriage is rebated to hold the test piece accurately, and is relieved in the neighbourhood of the slot so that the rubber is here gripped firmly.



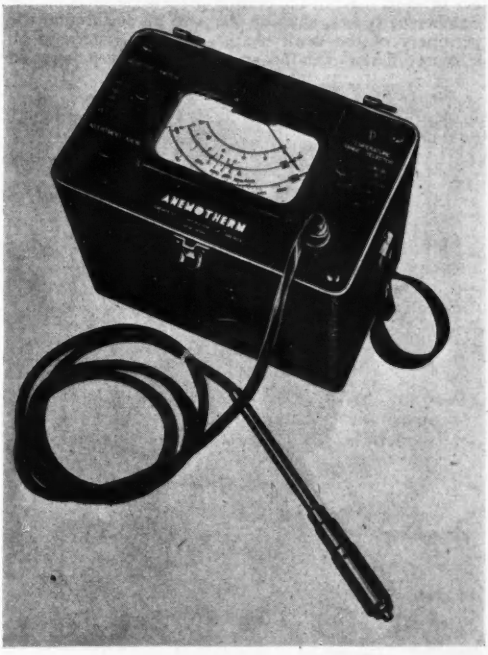
The height of the clamp which holds the razor blade is controlled by an adjustable screw top, the movement of which is measured by a dial micrometer reading in 1,000th inch. A simple lever raises the razor clamp against its stop and at the same time lowers two stops on the guide rails to permit the carriage to slide over the razor blade. The lever ensures that no cutting can take place until the blade is in position.

The test piece of sheeting is placed in the carriage and the lever is depressed to allow the carriage to travel over the blade which should first be brushed with soap solution. The carriage is moved back and forth six times between the stops, then withdrawn and the sample removed. The nick is measured by means of a microscope and the error taken up on the micrometer. Thus, if the nick is found to be 0·025-in, the razor blade clamp stop is lowered by 0·005-in measured on the micrometer dial, and the test repeated with a fresh sample.

THREE-WAY AIR METER

"Anemotherm" which gives air velocity, air temperature and static pressure readings at the turn of a knob, was recently placed on the market by the Anemostat Corporation of America, New York, N.Y.

Engineers, architects, contractors and installers will find the instrument invaluable for adjusting and testing equipment used for heating, ventilating and air conditioning. The probe is small and is attached to a long flexible cable—thus, readings are easily made anywhere. The new meter measures air velocity from 10 fpm to 5000 fpm and provides rapid-response measurement of temperatures from 30° F to 155° F. Either negative or positive static pressure may be read directly in inches of water, from ·05 to 10 positive and ·05 to 4 negative.



Because the instrument measures air velocity accurately regardless of direction of air flow, it can provide data on low air movement of a turbulent nature. Velocity readings are not influenced by any normal static pressure. The anemotherm is particularly valuable in measuring velocity and temperature of air confined in ducts, air currents in free spaces, and air entering or leaving through air outlets and inlets.

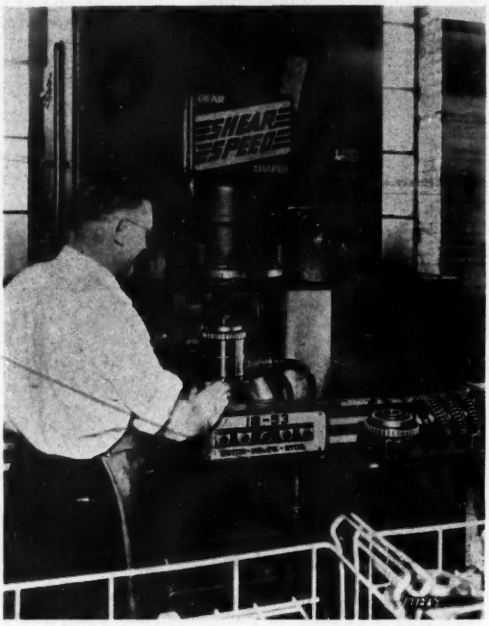
RUBBER HARDNESS GAUGE

The Clarkstan, Corp., Los Angeles, Calif., announce a new low cost accurate rubber gauge which measures the hardness of rubbers and other elastomers from 15

to 95 shore. Of convenient size, it may be carried in the pocket. The finish is of hard polished chromium. Each unit is supplied with a test block of known shore hardness.

"SHEAR SPEED" GEAR SHAPER

This interesting machine, developed by the Michigan Tool Co., to cut all teeth of a gear or external forms simultaneously, was demonstrated by Gaston E. Marbaix Ltd. at the recent Machine Tool and Engineering Exhibition. It has a capacity range of 3 to 5 inches, other models ranging from 1 to 10 inches. The number of tools used in the cutting process depends on the number of spaces, slots or forms to be cut around the periphery of the work blank. The tools may be all alike or different, and tool spacing may be even or varied

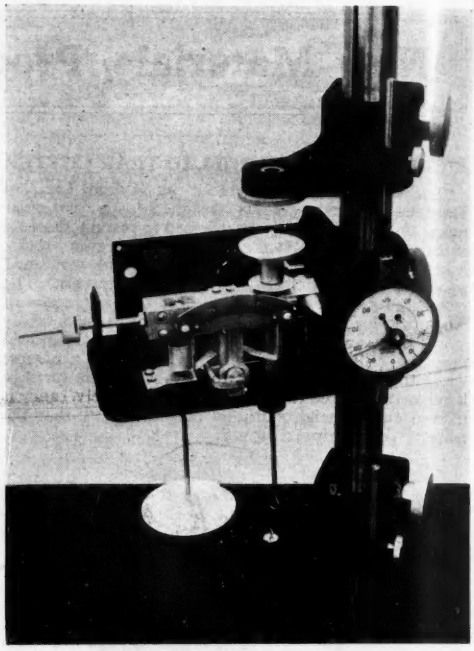


in accordance with the shape which is required to be produced. All the tools are fed radially into the work at the same time. As the cutting blades feed in, the length of work in contact with each cutting edge gradually increases. To balance the load on the machine and improve finishing accuracy, the feed is automatically decreased gradually as cutting proceeds. For example, the feed may be as much as 0·008-in for the initial stroke and as little as 0·0002-in for the final stroke. Operation is completely controlled by five electrical control buttons at the front of the machine.

COIL SPRING LOAD-TESTER

An improved load-tester for manufacturers and users of small springs, designed by the Hunter Pressed Steel Co., Lansdale, Pa., provides the higher speed and more accurate testing of coil springs which modern statistical inspection requires. To meet these demands, the new machine will test 50 to 200 springs with an accuracy of one part in 2,500 at maximum load, and do it in from 15 minutes to one hour. It will handle loads of up to 5 pounds, and spring lengths up to 12-in.

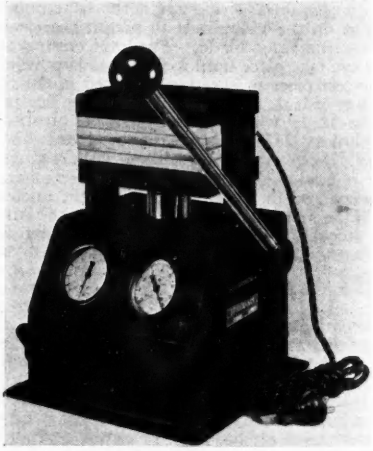
All compression-head surfaces are replaceable, and made of hardened steel which has been ground, lapped and chrome-plated. The tension hook is ball-mounted to swivel freely. An overload spring, mounted between the weighing beam and the weight pan, prevents damage from excess load or shock. The pointer mechanism, itself mounted in miniature ball bearings, amplifies beam motion 10 times for easy reading on a stainless steel scale graduated in 40 divisions. Limit of pointer



swing is adjusted by a conical beam-stop (which also can be screwed in further to lock the beam rigidly for protection during shipment). Red tolerance hands for go-no-go inspection are externally adjustable.

PORTABLE HYDRAULIC PRESS

A new compact hand-operated portable hydraulic press for laboratory or production work is introduced by Studebaker Machine Company, Maywood, Illinois. Optional with the unit are electrically heated platens for work requiring heat treatment.



Operation of the hand lever produces up to 8 tons pressure between platens. The new press can be efficiently adapted to broaching, assembling, piercing, oil grooving, riveting, sizing, forming, flanging, etc. Special dies can be developed for any type of product requiring press work.

CLASSIFIED ABSTRACTS

Numerous important abstracts which, for lack of space, cannot be published in our main editorial columns, are regularly included in condensed form in this section. Subscribers may obtain photostat copies of all original articles at cost.

INSULATING MATERIALS

Principles of Impregnation in the Electrical Industry

By J. NOVAK. (From *Elektrotechnicky Obzor, Czechoslovakia*, Vol. 37, No. 15, July 20, 1948, pp. 303-305.)

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(2) The immersion of the objects in the varnish should proceed in vacuo to prevent defects in the insulation. Subsequent application of pressure is not important.

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LIGHT METALS

Aluminium and Its Alloys

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METALLURGICAL ENGINEERING

The Determination of Non-Metallic Inclusions in Metals

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By A. H. WILSON BUSBY. (From *The Engineering Journal, Canada*, Vol. 31, No. 7, July, 1948, pp. 382-391, 19 illustrations.)

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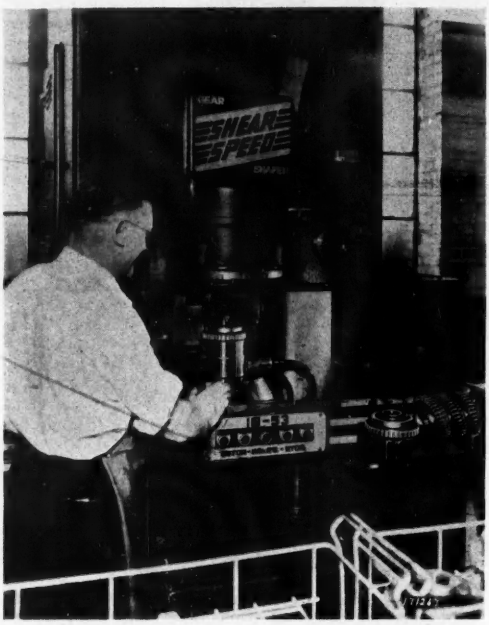
"SHEAR SPEED" GEAR SHAPER

This interesting machine, developed by the Michigan Tool Co., to cut all teeth of a gear or external forms simultaneously, was demonstrated by Gaston E. Marbaix Ltd. at the recent Machine Tool and Engineering Exhibition. It has a capacity range of 3 to 5 inches, other models ranging from 1 to 10 inches. The number of tools used in the cutting process depends on the number of spaces, slots or forms to be cut around the periphery of the work blank. The tools may be all alike or different, and tool spacing may be even or varied

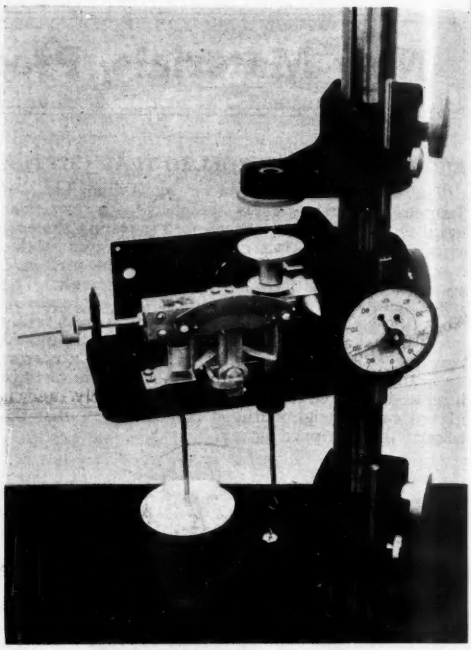


in accordance with the shape which is required to be produced. All the tools are fed radially into the work at the same time. As the cutting blades feed in, the length of work in contact with each cutting edge gradually increases. To balance the load on the machine and improve finishing accuracy, the feed is automatically decreased gradually as cutting proceeds. For example, the feed may be as much as 0·008-in for the initial stroke and as little as 0·0002-in for the final stroke. Operation is completely controlled by five electrical control buttons at the front of the machine.

COIL SPRING LOAD-TESTER

An improved load-tester for manufacturers and users of small springs, designed by the Hunter Pressed Steel Co., Lansdale, Pa., provides the higher speed and more accurate testing of coil springs which modern statistical inspection requires. To meet these demands, the new machine will test 50 to 200 springs with an accuracy of one part in 2,500 at maximum load, and do it in from 15 minutes to one hour. It will handle loads of up to 5 pounds, and spring lengths up to 12-in.

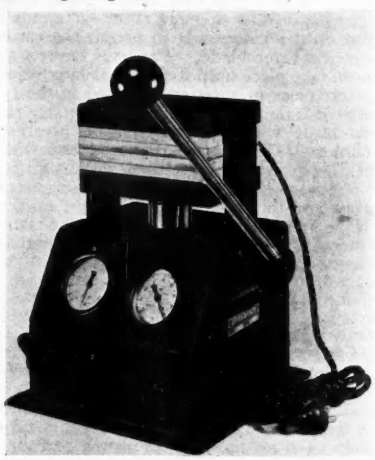
All compression-head surfaces are replaceable, and made of hardened steel which has been ground, lapped and chrome-plated. The tension hook is ball-mounted to swivel freely. An overload spring, mounted between the weighing beam and the weight pan, prevents damage from excess load or shock. The pointer mechanism, itself mounted in miniature ball bearings, amplifies beam motion 10 times for easy reading on a stainless steel scale graduated in 40 divisions. Limit of pointer



swing is adjusted by a conical beam-stop (which also can be screwed in further to lock the beam rigidly for protection during shipment). Red tolerance hands for go-no-go inspection are externally adjustable.

PORTABLE HYDRAULIC PRESS

A new compact hand-operated portable hydraulic press for laboratory or production work is introduced by Studebaker Machine Company, Maywood, Illinois. Optional with the unit are electrically heated platens for work requiring heat treatment.



Operation of the hand lever produces up to 8 tons pressure between platens. The new press can be efficiently adapted to broaching, assembling, piercing, oil grooving, riveting, sizing, forming, flanging, etc. Special dies can be developed for any type of product requiring press work.

CLASSIFIED ABSTRACTS

Numerous important abstracts which, for lack of space, cannot be published in our main editorial columns, are regularly included in condensed form in this section. Subscribers may obtain photostat copies of all original articles at cost.

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Pressure Vessels—*continued.*

vessel containing the catalyst used in the synthesis of ammonia. The vessel and the operating conditions are described in detail as well as the nature of the failure and the physical and metallographic examinations carried out to determine its cause. An examination of the steel indicated the formation of martensite, which caused cracking. Stresses are analysed, causes of the failure are discussed, and conclusions are arrived at. Finally some precautions necessary in the care and operation of high pressure vessels are listed.

SURFACE TREATMENT

Electrolytic Polishing of Copper in Orthophosphoric Acid

By R. W. K. HONEYCOMBE and R. R. HUGHAN. (From *Journal of The Council for Scientific and Industrial Research, Australia*, Vol. 20, No. 2, May, 1947, pp. 297-305, 5 illustrations.)

THE brightening of copper anodes in aqueous orthophosphoric acid electrolytes was investigated by measuring the anode potential with a calomel half cell. By plotting the current density for different anode potentials, a characteristic type of curve was obtained, the horizontal portion of which defined the range of conditions over which polishing occurred.

The variables investigated included temperature, concentration of the electrolyte, and the disposition of the electrodes. The theories of Jacquet and Elmore are discussed and are shown to be inadequate in explaining the experimental results.

An appendix discusses briefly the optimum conditions for the electrolytic polishing of copper specimens for metallographic examination.

WELDING

Welding Dissimilar Metals with Stainless Electrodes

By A. L. SCHAEFFLER. (From *The Iron Age, U.S.A.*, Vol. 162, No. 1, July 1, 1948, pp. 72-79, 10 illustrations.)

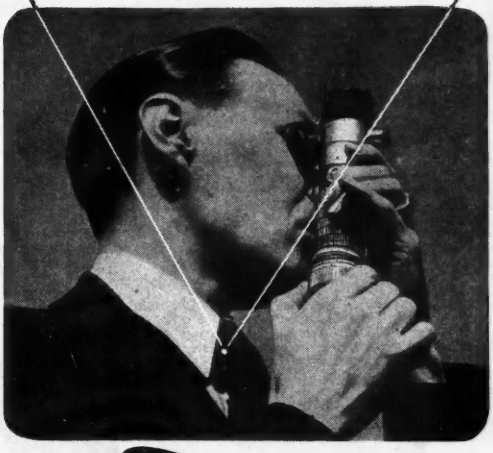
WITH the increase in number of different combinations of alloys and super-alloys being joined by metal arc welding, the fabricator finds it necessary to conduct a great deal of experimentation in order to obtain sound joints consistently. A graphical method is described in this article which makes possible the prediction of weld metal composition and structure, and embraces the joining of dissimilar metals utilizing single deposits and multi-pass welds. Based on the concept of dilution direction lines, this method employs the use of chromium and nickel equivalent values, making possible a consideration of most of the elements which affect microstructure.

THE USE OF OXYGEN IN STEELMAKING.

(*Continued from page 326*)

to temperature and silicon content and the use of the heat effect to hasten final scrap melting, the incidence of the lime boil, the speed of solution of ore into the oxidizing slag cover and the rapid heating of the bath to tapping temperature. This work has led to the conclusion that there is an immediate use for relatively small quantities of oxygen, directly applied to molten metal baths, for controlling the temperature of the slag-metal system and, thereby, the speed of the normal refining reactions, which will have wide application outside the steel industry proper. The use of oxygen in medicinal dosages to control the all important factor—bath temperature—may be the answer to many production problems in metals refining technology.

Accurate Light Measurement from 0·01 to 10,000 ft. Lamberts *on a spot this size*



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NEWS OF THE MONTH

PERSONAL

Mr. J. Belohoubek has commenced an industrial consultant's practice at "The Warren," Kendra Hall Road, South Croydon, Surrey. Telephone: Croydon 6964.

Major Frank Blackstone has been appointed a director of Blackstone & Co., Ltd.

Mr. J. W. Chaloner has been appointed assistant engineer in the Railway Department in Nigeria.

Mr. W. E. Cooper has been appointed assistant managing director (commercial), **Mr. F. J. Robinson,** assistant managing director (production), **Mr. G. Dring,** director of research and development, and **Mr. C. C. Last** and **Mr. G. W. Hodds,** joint general managers of Bakelite Ltd., 12 Hobart Place, London, S.W.1.

Mr. A. E. Emberton, J.P., has been appointed managing director of Peglers Ltd., Belmont Work, Doncaster.

Mr. W. M. B. Furniss has been appointed assistant managing director of the Electric Construction Co., Ltd.

Mr. G. C. I. Gardiner, chief designer of de Havilland Propellers Ltd., has been appointed chief engineer of the company. The design and development departments have been combined and **Mr. D. B. Russell** has been appointed chief designer. **Mr. C. C. Williams** has been appointed works manager.

Mr. N. A. Guttery has been appointed deputy secretary in the Ministry of Transport.

Mr. A. E. Higgs has been appointed a director of the National Smelting Co., Ltd.

Major J. Vivian Holman, A.F.R.Ae.S., M.I.Ae.S., has been appointed managing director of Philco (Overseas) Ltd. To free himself for his new duties, he has resigned his managing directorships of Philco Television and Radio Corporation of Great Britain Ltd., and Airmec International Sales Ltd., and his directorship of Airmec Laboratories Ltd., but retained his seat on the board of Radio and Television Trust Ltd.

Mr. F. H. Hoult has been appointed general manager of the light-castings department of Newton, Chambers & Co., Ltd., Thorncliffe, near Sheffield, to succeed **Mr. W. A. Curran** who resigned in order to take up engineering and foundry consultation work.

Sir Ellis Hunter, managing director of Dorman Long & Co., Ltd., has been elected chairman of the company.

Lieut.-General Sir W. G. Lindsell, G.B.E., K.C.B., D.S.O., M.C., who has been a director of Petrochemicals Ltd., for the past two years, has been elected chairman of the board in place of **Mr. H. Stuart Erben, O.B.E.** The latter has been chairman of Manchester Oil Refinery Ltd., since the inception of the firm in 1938 and retains his seat on the board of Petrochemicals Ltd.

Mr. C. Parry, Mr. D. F. Walton, Mr. W. Hall and **Mr. H. H. Mumby** have been appointed local directors of Thos. W. Ward Ltd., Sheffield.

Mr. A. J. Quig has been elected a deputy chairman of Imperial Chemical Industries Ltd.

Mr. K. B. Robinson, formerly of the steam turbine department of the English Electric Co., Ltd., has been appointed efficiency and testing engineer in the Yorkshire Division of the British Electricity Authority.

Mr. G. F. Shrigley has been appointed director and general manager of British Light Steel Pressings Ltd.

Mr. F. S. Strongman, A.M.Inst.Mun.E., has been appointed to the Colonial Service as a materials engineer in Kenya.

Mr. H. Tilsley has been appointed assistant export sales manager of The British Aluminium Co., Ltd., Salisbury House, London Wall, London, E.C.2. **Mr. A. E. Heeley** has been appointed manager of the Leeds branch office of the company in place of **Mr. W. H. Marston** who has been appointed to the London branch office.

Mr. W. E. Wood has been appointed executive engineer to the Public Works Department of Nigeria.

NATIONAL CONVENTION OF THE INSTITUTION OF PRODUCTION ENGINEERS

The first National Convention of the Institution of Production Engineers was held at Bournemouth from September 23rd to September 25th. More than three hundred members and many guests attended the Convention which was opened with a dinner at which the mayor of Bournemouth, Councillor J. W. Moore, welcomed the guests.

Addresses were given by the principal guest, General Sir Bernard Paget, on "Leadership"; by Dr. H. Schofield, President of the Institution, on "Industrial Relations"; by Mr. E. W. Hancock, on "Britain's Industrial Problems"; and by Lieut.-Gen. Sir Ronald Weeks, on "Britain's Industrial Future."

Discussion Groups dealt with important topics such as "Production Engineering" and "Engineering Production," under the chairmanship of Mr. A. McLeod; "Speculation and Enterprise as Factors in Planning for Repetitive Production," under the chairmanship of Mr. H. W. Bowen; "Youth in Engineering" and "Youth in Industry," under the chairmanship of Mr. E. W. Hancock; and "The Economics of Quality and Accuracy" under the chairmanship of Mr. E. P. Edwards.

The following Resolution was unanimously adopted and telegraphed to Rt. Hon. G. R. Strauss, Minister of Supply, signed by Mr. Herbert Schofield, President of the Institution:—

"This Convention of the Institution of Production Engineers, assembled at Bournemouth, and representing a large number of the country's senior production executives, deprecates the attitude of His Majesty's Government in failing to avail themselves, in the national interest, of the services of the Institution, and reaffirms the offer previously made to His Majesty's Government on behalf of the Institution by its council. Particularly now in view of the message received from the head of the United States Government Productivity division, namely, that 'the co-operation of the Institution of Production Engineers is vital in the work of the joint Anglo-American Productivity Committee,' the conference again assures His Majesty's Government of their full desire to co-operate, collectively, sectionally and individually, in every way possible."

"DESIGN AT WORK"

An Exhibition of Industrial Design, Burlington House, October 26th, 1948.

The first exhibition of the work of Royal Designers for Industry will be opened by H.R.H. The Duchess of Kent at the Royal Academy, 3 p.m., October 26th, 1948.

The exhibition, which is being sponsored by the Royal Society of Arts and the Council of Industrial Design, will be opened to the public from October 27th to November 28th. By explaining the designer's place in modern machine production and marketing methods, the exhibition will demonstrate his contribution to the manufacture of products that work well, look well, last well and sell well.

It is also intended that the exhibition shall be a useful forerunner to the centenary in 1951 of the Great Exhibition of 1851 by stimulating now the production of British goods of outstanding design in time for showing to the world during the 1951 Festival of Britain. In this connection, it is interesting to note that exactly 100 years ago, the Royal Society of Arts held one of a series of industrial exhibitions which led up to the Great Exhibition of 1851.

In a series of "case-histories" the work of the designer is studied in a number of subjects ranging from the production of a new radio set to the making of a wine glass, from the creation of a teapot to the design of a record breaking jet engined aircraft, from the shaping of an electric light fitting to the weaving of a utility fabric.

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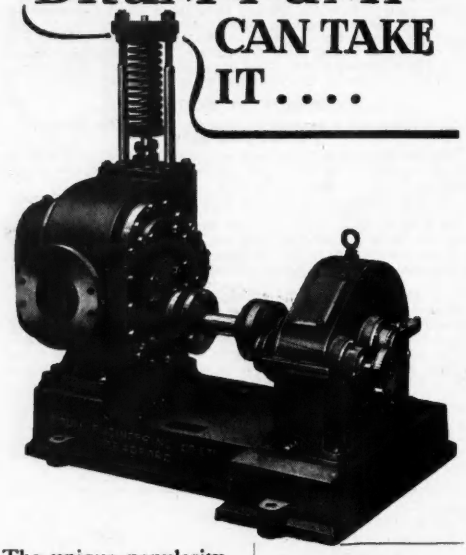

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"DRUM" Pumps will take any liquid, thick or thin. They can be run at high or low speeds. Their action is positive and affords a continuous flow, and their simple yet rugged construction makes the risk of breakdown negligible. In addition, "DRUM" Pumps are reversible in action, can be steam jacketed, and suit any form of drive.

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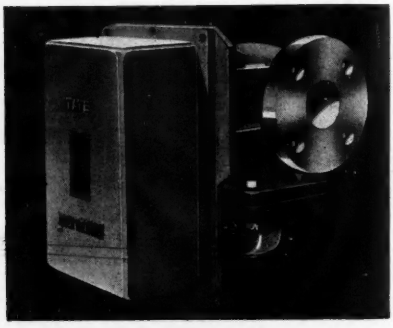

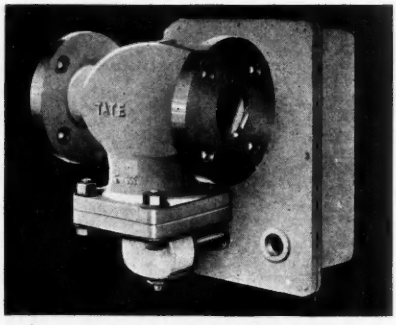

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Pressure Range: 5 lbs □" to 200 lbs □"

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BUSINESS NOTES.

Standard Wrapped Bushes introduced by the Glacier Metal Co., Ltd., of Alperton, Middlesex, in June, 1946, has been supplemented by 127 Metric sizes. In addition, the English sizes now include bushes with 2¼ in. and 2½ in. bores.

These butt-jointed bushes are made from steel lined with lead bronze. This bimetal combines high mechanical strength with good anti-friction properties and enables these bushes to be used in an endless variety of applications. For example, they are already well established in products ranging from diesel engines and fractional horse power motors to lawn mowers and plummer blocks.

These standard bushes are a new feature in engineering industry, where the advantages of standardisation become increasingly apparent. Greater scope is now provided by the Metric range and the additional English sizes.

Each size is mass produced so costs are low. Users therefore get the benefits of bulk production whether their requirements are small or large.

A chart of standard sizes can be obtained from the manufacturers.

E. H. Jones (Machine Tools) Ltd. of Edgware Road, The Hyde, London, N.W.9, regret to announce the closing down of their Halifax factory due to deterioration in the market which makes it impossible to continue production there on an economical basis of the types of machines manufactured at Halifax. The manufacture of these machine tools will in future be carried out in other works associated with the E.H.J. organisation.

INSTITUTION MEETINGS

The Institution of Mechanical Engineers

MEETINGS are held at the Institution, Storey's Gate, St. James's Park, London, S.W.1, at 6 p.m.

Oct. 22. General Meeting : President's Address.
Oct. 29. Joint Meeting with the Institution of Electrical Engineers : "Some Operating Experiences with High-Pressure Steam Power Plant," by W. N. C. Clinch.
Nov. 3 and Nov. 4. Lectures at 11 a.m., 2 p.m. and 4.30 p.m., on "Internal Combustion Turbines," by the staff of the National Gas Turbine Establishment, Farnborough.
Nov. 5. "On the Accuracy of Gear Hobbing Machine Tables," by J. M. Newton, B.Sc., M.I.Mech.E.
Nov. 9. Automobile Division General Meeting : "Stress-Carrying Rubber Components and their Vehicle Applications," by S. W. Marsh.
Nov. 12. Applied Mechanics Group : "Some Properties of the Compression Shock as in Turbine and Compressor Blade Passages," by I. M. Davidson, B.Sc.
Nov. 19. Thomas Hawksley Lecture : "Heat Engines," by K. Baumann, M.I.Mech.E.
Nov. 26. "Modernization of a Coal Discharging Depot," by G. T. Shoosmith, M.A., M.I.Mech.E.
Dec. 3. "Control of Production," by S. W. Lister, M.A., B.Sc. (Eng.), A.M.I.Mech.E.
Dec. 10. Applied Mechanics Group : "General Review of Fluid Flow through Granular Beds," based on Papers by Dr. H. E. Rose, M.Sc. (Eng.), A.M.I.Mech.E., and Dr. A. M. A. Rizk, B.Sc. (Eng.), G.I.Mech.E.
Dec. 14. Automobile Division General Meeting : "Lubrication of Engines in Public Service Vehicles," by A. T. Wilford, B.Sc.
Dec. 17. "Cinematography in Engineering," by H. A. V. Bulleid, M.A., A.M.I.Mech.E.

The Institute of Marine Engineers.

Meetings will be held at the Institute, 85 The Minories, London, E.C.3, at 5.30 p.m.

Nov. 9. "Metal Spraying in Relation to Marine Engineering," by J. Barrington Styles, M.I.Mech.E.
Dec. 14. "Latest Developments in Reversible Propellers," by Professor L. C. Burrill, M.Sc., Ph.D.

The Institute of Welding—North London Branch.

All Meetings commence at 7.30 p.m.

Nov. 18. "Welding in Constructional Engineering," by O. Bondy, A.M.I.C.E., M.Inst.Struct.E., at the Polytechnic, Regent Street.
Dec. 15. "Open Discussion on Welding Methods," at East Ham Technical College.
Jan. 19. Presidential Address at the Polytechnic, Regent Street.
Feb. 16. "Conveyors," by A. T. C. Burrows, B.Sc., A.M.I.Mech.E., at Acton Technical College.
Mar. 16. "Developments in the Technique of Resistance Welding," by C. A. Burton, at the South West Essex Technical College, Walthamstow.
Apr. 13. Annual General Meeting at the Polytechnic, Regent Street.

Society of Instrument Technology—Midland Section.

Meetings will be held at the Arden Hotel, New Street, Birmingham, at 7 p.m.

Oct. 22. "Continuous Gauging of Rolled Materials," by F. C. Butcher, of Taylor, Taylor & Hobson Ltd.
Nov. 19. "Thermocouples," An Introduction to a discussion by Eric E. Hancox, of Integra Co., Ltd.
Dec. 17. "Optical Devices," by E. Simpson, A.I.R.I., of the Dunlop Rubber Co.
Jan. 21. "Impressions of recent American Instrumentation," by A. J. Young, read by G. P. Clay, of I.C.I. Ltd.

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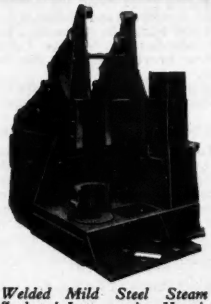

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